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of the

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OCTOBER, 1948

VOL. 70, NO. 7

Transactions

of The American Society of Mechanical Engineers

Published on the tenth of every month, except March, June, September, and December

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Published monthly by The American Society of Mechanical Engineers. Publication office at 20th and Northampton Streets, Easton, Pa. The editorial department is located at the headquarters of the Society, 29 West Thirty-Ninth Street, New York 18, N. Y. Cable address, "Dynamic," New York. Price, \$1.50 a copy, \$12.00 a year; to members and affiliates, \$1.00 a copy, \$6.00 a year. Changes of address must be received at Society headquarters three weeks before they are to be effective on the mailing list. Please send old as well as new address. . . . By-Law: The Society shall not be responsible for statements or opinions advanced in papers or . . . printed in its publications (B13, Par. 4). . . . Entered as second-class matter March 2, 1928, at the Post Office at Easton, Pa., under the Act of August 24, 1912. . . . Copyrighted, 1948, by The American Society of Mechanical Engineers. Reprints from this publication may be made on condition that full credit be given the Transactions of the ASME and the author, and that date of publication be stated.

Development of Cleavage Fractures in Mild Steels

By A. B. BAGSAR,¹ PHILADELPHIA, PA.

The susceptibility of several types and thicknesses of mild steel of ship-plate and pressure-vessel qualities and of samples of welds to development of cleavage or brittle fractures has been determined by a new test, termed the "cleavage-tear" test, in which a notched tensile-bend type of test coupon is used. The effects of notch and coupon geometries, load eccentricity, rate of loading, testing temperature, and of heat-treatments were investigated.

At temperatures below the transition-temperature range, the presence of a notch of proper geometry and orientation was found to create a state of multiaxial stress which appears to expand the elastic range of steel and to raise its proportional limit almost to coincide with its breaking point. The breaking strength of rectangular sections containing a notch of the foregoing type was found to decrease, and susceptibility to cleavage fracturing to increase as the notch root radius was decreased. Within the limits investigated, the breaking strength of steel was found to be modified by the notch geometry, this modification being practically independent of the size effect if the section was 6 in. or greater in depth. The transition temperatures for the steels investigated in the as-rolled condition were found to be about 100 deg F higher than those indicated by the conventional Charpy impact test, and are considered to be more nearly indicative of behavior of steels in service. Nevertheless, the transition temperature of a steel cannot be defined for service conditions unless the state of stress which prevails under the same conditions is also defined.

Two types of fracture were encountered, one being the brittle or cleavage type exhibiting no appreciable deformation and lower nominal breaking strength, and the second the shear type exhibiting normal ductility and higher breaking strength. Fractures above the transition range were of the shear type and below that temperature range were of the cleavage type. On the basis of the test data presented, reconsideration of our present factors of safety is suggested for monolithic structures with the object of safeguarding against the possibility of development of large cleavage fractures from small discontinuities, notches, or cracks. Other remedial measures for minimizing the damage of cleavage fractures, including modifications in design and material of construction, are suggested.

INTRODUCTION

NUMEROUS failures in all-welded ships, pressure vessels, and structures through development of cleavage or brittle fractures received considerable publicity during World War II and attracted earnest attention. Steels manifesting satisfactory strength and normal ductility, as determined by the

conventional tensile and bend tests, were found to fail in certain services by cleavage fractures, exhibiting no evidence of any appreciable ductility. In view of the foregoing, a number of research programs were inaugurated by various governmental agencies and private laboratories to investigate this phenomenon and search for remedial measures (1).² Much has been learned and much remains to be learned about cleavage fractures. An understanding of this phenomenon and its interpretation are of importance to all industries fabricating or using metals, including the petroleum industry, whose production, processing, and distribution facilities depend on metals, mostly on steel.

In this paper are presented some of the results of a research undertaken about 4 years ago, with the object of studying cleavage or brittle fractures in steels of ship-plate and pressure-vessel qualities. This work was modest in its scope at the beginning but became more involved as it progressed, due to complications introduced by numerous variables affecting development of such fractures. It was pursued as actively during the war years as possible, although it was almost suspended several times as other more urgent work intervened. This research was done under the direction of the author by the Metallurgical Section of the Sun Oil Company, most of the steels for the tests and some of the testing facilities being furnished by the Sun Shipbuilding and Dry Dock Company.

A search in the literature revealed no comprehensive or standard testing procedure for investigating cleavage fractures or for a rational study of the numerous factors and variables affecting development of this type of fractures. Therefore a new approach to this problem and development of a new testing method became necessary.

Cleavage fractures develop in steel plates and structural members of rectangular or complicated cross section, by multiaxial and unsymmetrical stress systems, imposed by various service conditions. It is obvious that the test coupon used had to be of a type which would simulate at least some of the basic conditions of loading or stressing encountered in service. In the conventional tensile tests, the imposed stress is unidirectional. In the notched cylindrical tensile coupon, although lateral stresses are introduced by the presence of the notch, the cross section of the coupon is symmetrical. The notched-bar coupon used in impact tests is profoundly affected by local nonhomogeneity of the material tested and by the "mass effect," and yields qualitative rather than quantitative results. Hence none of the foregoing test coupons and testing methods were considered suitable for this investigation.

Exploratory tests revealed that a notched-bend coupon of large dimensions (2) might be suitable for this investigation. This coupon consists of a notched simple beam positioned edge-wise and loaded in the center of the unnotched edge, with supports near the ends of the opposite edge. However, some important limitations of this test coupon were soon discovered. These included the difficulty of avoiding the severe distortion and, in some cases, buckling of the coupon at the points of support and of load application, also the difficulty of computing the stresses involved.

² Numbers in parentheses refer to Bibliography at end of paper.

¹ Chief Metallurgical Engineer, Sun Oil Company.
Contributed by the Metals Engineering Division and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-75.

A new and much more satisfactory type of test coupon was developed and used in this investigation. It is a tensile-bend type of coupon, which can be considered as being a notched beam of rectangular cross section subjected to combined axial tensile and transverse bending stresses, induced by eccentric loading. We shall call this the "cleavage-tear" test coupon, which term is the most descriptive one. Its details are described in the next section of this paper. This coupon was adopted for this investigation because it simulates some of the basic conditions of loading encountered in service and also because it is suitable for determining quantitatively the effect on the development of cleavage fractures of a number of important variables such as eccentricity of loading, coupon depth, notch depth, notch angle, radius at the base of the notch, and several other variables. This coupon has the further advantage of enabling one to test relatively large plate sections in full thickness, with a minimum amount of machining work.

It could be stated in simple terms that cleavage fractures develop by separation of the metal along certain cleavage planes, whereas the shear fractures develop by sliding of one part over the other to the point where separation or failure occurs. It is possible to produce, in a given section of steel, cleavage fractures with considerably lower loads than the loads required to produce shear fractures. On the other hand, by application of a given load, it is possible to produce cleavage cracks in steel whose cross-sectional area is many times greater than that affected by shear fracturing by the same load. It is endeavored to show by this paper that failure of steel by cleavage fractures conforms to some laws and modified equations, and that conditions which favor development of such fractures can be evaluated reasonably accurately, even if some empirical terms are used to compensate for the effect of certain variables. These empirical terms could probably be reduced to rational terms as additional data become available. It is also endeavored to indicate by the paper certain design and material-specification details by means of which the occurrence of cleavage failures and the extent of damage caused by them could be minimized.

TESTING METHOD AND PROCEDURE

The Test Coupon. The test coupon used is essentially a short beam of rectangular cross section, which is notched at its edge and subjected to tensile loading, eccentrically applied. Its details are shown in Fig. 1 for various eccentricities of loading and coupon depths. Figs. 2 (a) and 2 (b) show 1-in-thick steel coupons of 6 and 22 in. depths, respectively, after they were broken in the test, developing cleavage fractures.

It is to be noted in Fig. 1 that the coupon has a U-shaped recess on the top edge, the upper parts of the arms accommodating loading holes and providing the desired eccentricity of loading. The coupon is notched at the center of the slot, so as to induce a state of multiaxial stress when loaded. The tensile load is applied through a suitable clevis and pin arrangement, the pins being inserted in the round loading holes of the test coupon. The holes are reinforced if the total load required to cause fracturing produces excessive deformation in the holes. This method of load application being of "hinged" type was found to be very satisfactory, since it practically eliminates auxiliary stresses induced by end restraints throughout the entire cycle of loading, and of fracture propagation.

In order to obtain the desired rigidity and to prevent excessive deflection or failure along the arms and sides, proper ratios of width to depth of the coupon had to be used. It was also necessary to use a proper width for the slot so as to prevent end effects at the bottom of the slot and excessive deflection or buckling along the depth of the coupon. The dimensions shown in Fig. 1 were chosen with the foregoing in view. In all

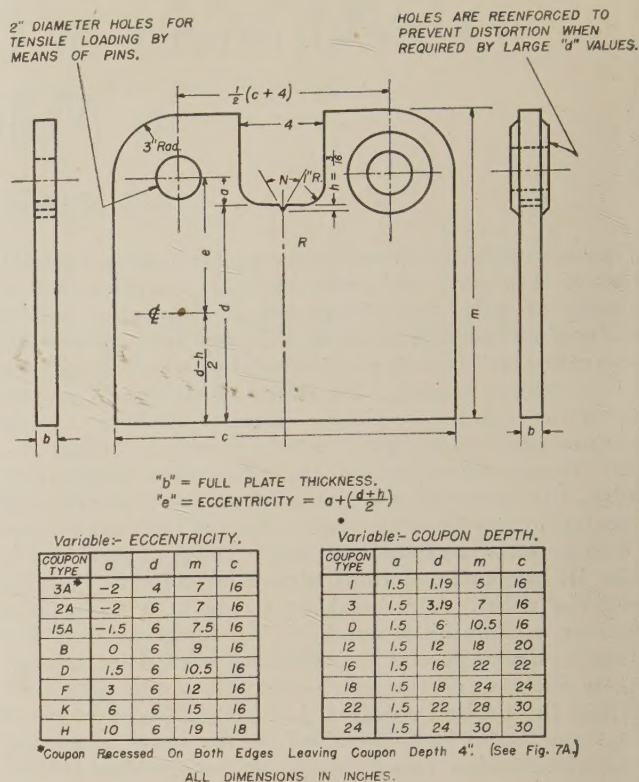


FIG. 1 DETAILS OF CLEAVAGE-TEAR TEST COUPONS FOR VARIOUS ECCENTRICITIES OF LOADING AND COUPON DEPTHS

cases the thickness of the test coupons used was the full thickness of the steel plate which was to be tested.

The test coupons were rather easily prepared by cutting the plate with a torch to the proper size and shape, and machining to a depth of about $\frac{1}{4}$ in. the face that was to be notched so as to remove the metal affected by the heat of torch cutting, before notching was done. The weld-test coupons were made by butt-welding 1-in-thick plates together, from which D-type coupons, Fig. 1, were cut in such a manner as to locate the weld seam on the vertical axis of the coupon. By this arrangement the notch was located in the weld-deposited metal, and the fracture occurred in the latter when tested.

It is apparent from the geometry of the test coupon used that, as a tensile load is applied, two types of tensile stress are developed at the slotted edge, one being the axial stress produced by the direct tensile load, and the other being the resultant tensile stress produced by bending. Assuming that no notch is present and that the loading is such as to produce no excessive deflection or curved-beam conditions, the total stress s can be found within the elastic range by superposing the two stresses mentioned, as follows

$$s = \frac{P}{A} + \frac{Mc}{I} \dots \dots \dots [1]$$

in which P/A is the applied tensile load in pounds divided by the cross-sectional area in square inches, M is the maximum bending moment in inch-pounds due to load P , c is the distance in inches from the outer fiber to the neutral axis and I is the moment of inertia of the cross section with respect to the neutral axis.

At the bottom edge of the coupon the flexural stress is compressive, and therefore the net stress at that edge will be as follows

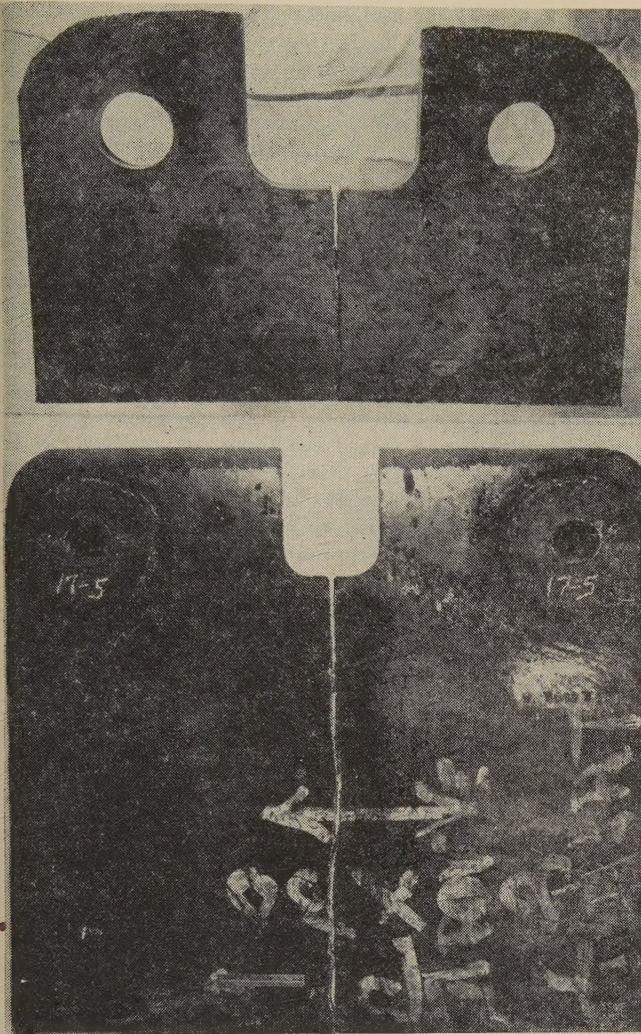


FIG. 2 (a) (Top) CLEAVAGE-TEAR COUPON WITH 60-DEG NOTCH, 6 IN. DEPTH AND 4½ IN. ECCENTRICITY OF LOADING, AFTER TESTING AT 85 F
(Note the cleavage fracture developed.)

FIG. 2 (b) (Bottom) CLEAVAGE-TEAR COUPON WITH 45-DEG NOTCH, 22 IN. DEPTH AND 13½ IN. ECCENTRICITY OF LOADING (Broke into two pieces as cleavage fracture penetrated entire depth of coupon when tested at 32 F.)

$$s = \frac{P}{A} - \frac{Mc}{I} \dots \dots \dots [2]$$

If a notch is introduced, and the tensile load is increased to the breaking point of the steel, reconsideration of the foregoing stress analysis becomes necessary. By introducing a proper notch at the edge of the test coupon, a multiaxial state of stress is created, by virtue of which the yield load of the system is raised almost to coincide with the breaking load, as the experimental data presented in this paper show, and the total deflection produced before fracturing is negligible. Under the circumstances, Equation [1] can be assumed to be applicable, if certain correction factors are introduced to compensate for the multiaxial state of stress and for the stress distribution which prevails in the presence of a notch. It is obvious that Equation [2] need not be considered under conditions of fracturing, since as the fracture occurs the neutral axis is progressively depressed, and the stress at the bottom edge is materially altered.

Experimental data indicated that in so far as establishing a

relationship between the breaking loads and the section and notch geometries is concerned, it would not be necessary to evaluate the stress-distribution details at the notch, if the proper size and type of coupons were used. It was found that if the section were considered as a whole, the P/A value required to produce fracturing would represent the integral of all the resultant loads acting on the entire section subjected to fracturing, and that the following relationship would be valid

$$S_N = \frac{P}{A} + nq \left(\frac{Mc}{I} \right) \dots \dots \dots [3]$$

where

S_N = nominal breaking strength, psi. For cleavage fractures, S_N = nominal cleavage strength, S , psi. For the latter type of fractures S_N can also be considered as being the nominal unit breaking stress

P = applied tensile load, lb, required to cause fracturing

A = cross-sectional area, sq in., subjected to fracturing = $(d - h)b$

b = thickness of coupon, in. = plate thickness

d = depth of coupon in unnotched section, in.

h = depth of notch, in.

M = bending moment, in-lb = Pe

I = moment of inertia of section, in.⁴ = $(d - h)^3b/12$

e = load eccentricity, in., measured from centroid (Fig. 1), to center of loading hole = $a + (d + h)/2$

c = distance of bottom of notch from centroidal axis = $(d - h)/2$ in.

n = correction factor

q = anisotropy factor, arbitrarily assumed to equal 1 for fractures transverse to direction of rolling of plate

If the fracture developed is of the ductile or shear type, Equation [3] should be written in the following form

$$S' = \frac{P}{A} + nqv \left(\frac{Mc}{I} \right) \dots \dots \dots [4]$$

in which S' is the nominal shear strength of the steel, psi, under tear conditions and in the presence of a notch, v is a variable correction factor which includes the curved-beam factor, required by excessive deflection caused by shearing. The other symbols have the same meaning as in Equation [3]. Fractures which occur above the transition temperatures are normally of the shear type even in the presence of a sharp notch and, as will be shown in the subsequent sections of the paper, require considerably higher fracturing loads than those required by cleavage fractures originating from a notch of the same geometry but below the transition temperatures, that is, S' normally has a higher numerical value than S .

If a section of steel plate is broken under such test conditions that the fracture obtained is of brittle or cleavage type and this fracture is transverse to the direction of rolling of the plate, the factor q can be dropped from Equation [3]. The latter equation, by substitution of the foregoing values, can then be put in the following form

$$\frac{P}{A} = \frac{S}{1 + n \left(\frac{6e}{d - h} \right)} \dots \dots \dots [5]$$

Equation [5] was found to be valid for evaluating the breaking loads for transverse sections, except additional modifications are necessary to compensate for a number of other variables affecting breaking loads. It is obvious that in Equation [5] P/A represents the applied unit load to produce fracturing, and not unit

stress at the notch. It is also obvious that the actual unit stress at the notch cannot be evaluated by Equation [1] or [5], due to disturbance by the notch of the normal stress distribution, unless the stress-concentration factor and other necessary details are known.

The term P/A , designated as the unit breaking load, is used throughout the paper as the index for comparing the data obtained under diverse conditions of testing. The advantage of such an index consists in the fact that both P and A are measured directly by tests for each condition, without making any assumptions. However, it is apparent that P/A is not a fixed load in cleavage-tear tests unless the coupon and notch geometries and other variables are fixed, as indicated by Equation [5]. Therefore it was necessary to use certain standard types of coupon and test conditions for determining the effect of a single variable in each series of tests.

Stress measurements and stress patterns appear to show that if the extent of the over-all deflection produced is slight, that is, if the fracture which occurs is of the cleavage type, the distance y between the neutral axis and the centroid for the standard D-type coupon, Fig. 1, is approximately as follows

$$y = -\frac{(d-h)^2}{12e} \dots \dots \dots [6]$$

The neutral axis for this is then y in. below the centroid. It appears then that the position of the neutral axis of a notched bar is about the same as the unnotched bar, and that it is lowered by a distance approximately equal to the depth of the notch, if the section is 6 in. in depth. Experimental evidence shows that in notched sections with depths greater than 6 in., the effective eccentricity of loading is approximately the same as that for the 6-in. section, if the loading eccentricity component a , Fig. 1, is the same.

In case of cleavage fractures, apparently no appreciable shifting of the neutral axis occurs from the foregoing positions even though some local yielding at and near the notch occurs as the load levels approach the breaking point. Experimental data seem to indicate that the factor n compensates fairly effectively for the assumptions made in deriving Equation [5].

According to Equation [6] it would appear possible to arrive at such ratios of coupon depth $(d-h)$, to loading eccentricity e , that the entire section of the coupon under the notch will be subject only to tensile stress, the axis of zero stress in such a case being lowered to or below the lower edge of the coupon. The upper or notched edge will then have the maximum tensile stress. The stress will decrease to lower values or zero as the bottom edge of the coupon is reached, depending upon the values of $(d-h)$ and e selected.

Preparation of Notch. A special cold-pressing method of notching was developed and used in these tests. This consisted of pressing a knife-edged die block of a given angularity and sharpness, made of heat-treated high-speed steel, into the edge of the test coupon, with suitable guide jigs to keep the notch in its proper location and to the predetermined depth. The pressing of the notch was done in the compression section of a hydraulic tensile testing machine. It was found that, generally, more reproducible results and sharper notch radii could be obtained by this method than by ordinary methods of machining. To obtain this result, however, it was found necessary that the knife-edge of the die block be prepared by careful grinding, checked periodically under the microscope as it is used, and reground as required in order to keep it in its original sharpness, angularity, and root radius.

Numerous exploratory tests showed that in these tests the entire section of the coupon subjected to tearing or fracturing

determines the breaking load, and that these tests are relatively free from the influence of local irregularities which do not change the notch angularity and particularly the radius at the base of the notch. The local cold-working and slight burring at the edges of the notch, produced by the method of cold-pressing, were found by numerous tests to have no appreciable effect on the net breaking load of the section or on the fractures obtained. Considerable deformation or yielding has been observed to occur in all notches, especially in those with large root radii before fracturing occurs.

In Figs. 3 (a) and 3 (b) are shown, at a magnification of 50 diam, sections through 45 and 60-deg notches with 0.003 and 0.0015 in. root radii, respectively, prepared by this method. Most of the notches used in this investigation were 45-deg cold-pressed notches with a root radius of 0.0015 in., sections of which notch are shown in Figs. 3 (c) and 3 (d) at a magnification of 100 diam, in unetched and etched conditions, respectively. It is to be noted from Fig. 3 (d) that the depth of penetration of cold-working is rather shallow at the base of the notch, although it is appreciable along the side walls of the notch. The dimensions of the notch radii were controlled by the shape of the knife-edge of the die.

Testing Procedure. All tests were made on a 300,000-lb-capacity hydraulic tensile testing machine, with the use of suitable clevis and pin-type grips for applying the tensile load. Unless otherwise stated, the rate of load application was 2 ipm in all tests, which is about the maximum crosshead separation speed of the machine used. The test coupons, if tested at temperatures other than ambient, were kept for at least 1 hr, in a bath of suitable capacity held at the prescribed temperature, before subjecting them to the cleavage-tear tests.

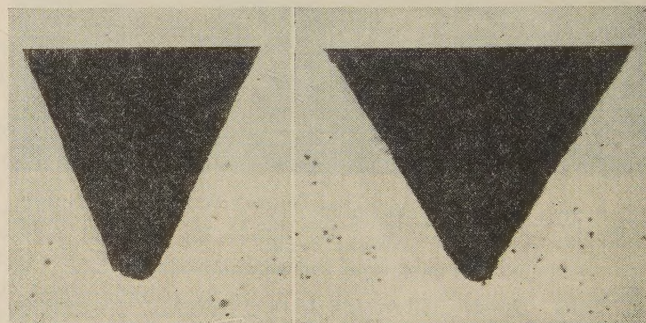


FIG. 3 (a)

FIG. 3 (b)

(Figures a and b are sections through 45 and 60-deg notches with 0.003 and 0.0015 in. root radius, respectively, formed by cold-pressing; $\times 50$.)

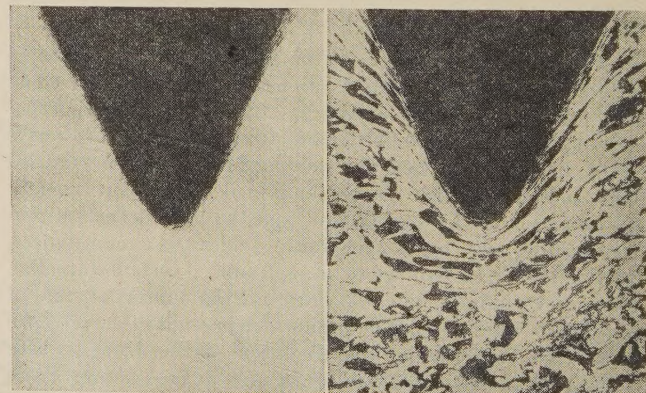


FIG. 3 (c)

FIG. 3 (d)

(Figures c and d show in unetched and etched condition, respectively, a section through a 45-deg notch of 0.0015 in. root radius; $\times 100$.)

In evaluating the notch sensitivity of steels it has been the general practice to measure the energy absorbed or the energy spent to effect the fracture. This is the practice used in impact tests. In some other tests the specific work of fracturing is used as a measure or index of resistance to fracturing. The measurement of fracture work was found in most cases to be inapplicable to the cleavage-tear tests. A much simpler index was found for evaluating the resistance or susceptibility of a given steel to development of cleavage fractures, in these tests, this index being the P/A values as has been previously stated.

Exploratory tests summarized in Table 1, run in the early stages of this investigation, indicated that the breaking load might be such an index. In these, $3\frac{1}{2}$ -in.-deep simple beam coupons were used, with a notch depth of $\frac{3}{16}$ in. The load was applied at the center of the upper edge of the beam, opposite to the edge containing the notch, the beam being supported on round pins near the two ends of its lower edge. By this loading the notched edge of the coupon was in tension, and the opposite edge in compression. Steel 7, Table 2, in the as-rolled condition was used for these tests. For comparison, three coupons containing notches of various sharpness at the base and one unnotched coupon were used to determine the relation between the maximum deflection at the center of the beam at the time of fracturing and the load required to break the beam. The three types of notch used were a 45-deg V-shaped pressed notch with 0.003 in. radius at the base, the standard Izod 45-deg V-machined notch with 0.01-in. notch radius, the third being a standard Charpy keyhole notch with 0.039 in. root radius. The pressed

notch with radius of 0.003 in. showed the least deflection and the least breaking load. The beam containing the Izod notch of 0.01 in. radius required higher breaking load and showed greater deflection. The Charpy notch required a still higher breaking load and produced a greater deflection than the two notches mentioned. No fracture was produced in the unnotched beam even after application of a high load and a bending deflection of $4\frac{1}{2}$ in.

The relationships between the breaking load and the deflection, deflection and the notch radius, and between the fracture-work index and the notch radius are graphically presented in Fig. 4 on log-log co-ordinates. Additional test data are given on the relationship between the unit breaking loads and notch radii in a subsequent section of the paper. These data show that under cleavage-tear test conditions, the unit breaking load indicates with sufficient accuracy the resistance of a steel to fracturing, higher breaking loads being required if greater amounts of flexing or deformation are to occur.

MATERIALS OF TEST

Eight steels were used in this investigation, varying in thickness from $\frac{1}{2}$ to $2\frac{1}{4}$ in., the chemical composition and the physical properties of which are listed in Table 2. Several exploratory tests were also made on mild-steel butt-welded seams in 1-in.-thick plates welded by the manual- and submerged-arc processes. The latter were accepted for the tests on basis of x-ray inspection and were not tested for chemical or physical properties.

It is to be noted that all the steels have a normal chemical

TABLE 1 RELATIONSHIPS OF BREAKING LOAD P , DEFLECTION F , NOTCH RADIUS R , AND INDEX OF FRACTURE WORK, W

Test Coupon: Simple beam, 1 in. thick $\times$ $3\frac{1}{2}$ in. deep $\times$ 16-in. span, with $\frac{3}{16}$ -in. deep notch
All tests were made at 75 F, and with a loading speed of 0.3–0.5 in./min.

Type of Notch	Notch radius, R , in.	Breaking load, P , lb	Deflection, F , in.	Work index $W = P \times F$, in-lb	Initial fracture
Pressed 45° V	0.003	33,400	0.06	1990	Cleavage
Pressed 45° V	0.003	33,000	...	...	Cleavage
Average	0.003	33,200	0.06	1990	
Machined Izod 45° V	0.010	39,200	...	...	Cleavage
Machined Izod 45° V	0.010	37,800	0.16	6160	Cleavage
Average	0.010	38,500	0.16	6160	
Standard Charpy, keyhole	0.039	44,800	...	...	Cleavage-shear
Standard Charpy, keyhole	0.039	46,400	0.44	20,060	Cleavage
Average	0.039	45,600	0.44	20,060	
Coupon not notched	...	(63,400)	4.125	261,500	No fracture

TABLE 2 CHEMICAL AND PHYSICAL PROPERTIES OF STEELS TESTED

Standard Rectangular Test Coupons with 8-In. Gage Length, Conforming to ASTM Specification E8, Were Used

	Steel 7	Steel 12	Steel 13	Steel 15	Steel 16	Steel 17	Steel 20	Steel 22
Plate thickness, in.	1	$\frac{3}{4}$	$1\frac{1}{4}$	1	1	1	$2\frac{1}{4}$	$\frac{1}{2}$
Chemical composition								
Carbon, %	0.21	0.33	0.26	0.20	0.17	0.19	0.18	0.20
Manganese, %	0.44	0.55	0.46	0.45	0.40	0.42	0.44	0.45
Phosphorus, %	0.020	0.014	0.015	0.019	0.012	0.015	0.014	0.020
Sulphur, %	0.033	0.035	0.030	0.029	0.034	0.030	0.026	0.045
Silicon, %	0.01	0.02	0.01	0.07	0.18	0.07	0.20	0.01
Aluminum, %	...	...	...	...	0.044	...	...	...
Titanium, %	...	...	...	...	0.006	...	...	...
Tensile properties								
Longitudinal tensile (fracture Transverse to Rolling)								
Tensile strength, psi.	60,100	60,500	57,150	59,300	64,500	58,000	62,000	60,500
Yield point, psi.	32,900	38,600	31,800	30,400	39,300	32,900	39,000	37,000
Elongation, 8 in., %	30.7	30.3	30.0	31.0	28.0	31.0	33.0	27.0
Transverse tensile (fracture Parallel to Rolling)								
Tensile strength, psi.	58,400	60,500	58,200	58,500	...	56,000	...	...
Yield point, psi.	31,200	35,200	30,200	31,400	...	32,400	...	...
Elongation, 8 in., %	32.2	29.6	31.0	31.3	...	27.0	...	...

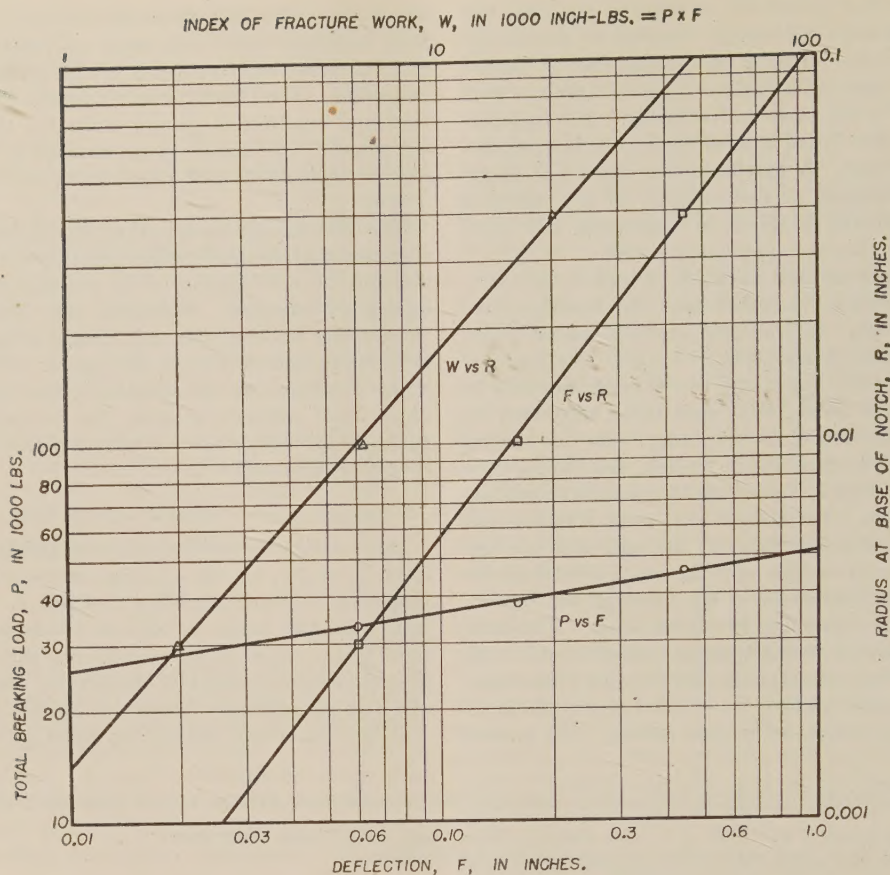


FIG. 4 RELATIONSHIP BETWEEN BREAKING LOAD AND DEFLECTION, NOTCH RADIUS AND DEFLECTION, AND BETWEEN RADIUS AT BASE OF NOTCH AND INDEX OF FRACTURE WORK

composition, and that their tensile strengths are within the range of 56,000–60,500 psi, with an average elongation of about 30 per cent in 8 in., except the killed steels 16 and 20, which show a somewhat higher tensile strength and yield point.

The microstructures of representative steels 7, 15, 17, 16, 20, and 22 in the as-received condition are shown in Figs. 5 (a) to 5 (f), inclusive, at a magnification of 100 diam. As is noted, steels 7, 15, and 22 show distinct segregation banding and are ship-plate quality. Steels 7 and 22 are rimmed steels, and 15 is semikilled. Steel 17 is less banded and is also of semikilled quality. The killed steels 16 and 20 show practically no banding.

The austenitic and actual grain sizes, McQuaid-Ehn normality (ASTM E 19-46), and the inclusion content (ASTM E 45-48T) of each steel are listed in Table 3.

TABLE 3 GRAIN SIZES, NORMALITIES, AND INCLUSION CONTENTS OF STEELS TESTED

Steel no.	Grain size		Normality	Inclusions
	Actual	Austenitic		
7	6-7	2 1/2	Normal	D thin 4 and C heavy 3
12	7	2 1/2	Normal	D thin 5 and C thin 5
13	6	3	Slightly abnormal	D heavy 4 and C heavy 3
15	6	3	Normal	D thin 4 and D heavy 3
16	8	8 1/2	Slightly abnormal	D thin 4 and D heavy 4
17	6	3	Normal	D thin 5 and A heavy 3
20	5-6	5-6	Normal	D heavy 5 and C thin 4
22	8-9	5-6	Normal	D thin 4 and A heavy 2

The microstructure of welds was found to be satisfactory and comparable to that of welds meeting the requirements of Paragraph U-68 of the ASME Code for Unfired Pressure Vessels.

DETERMINATION OF EFFECT OF VARIABLES ON DEVELOPMENT OF CLEAVAGE FRACTURES

The behavior of steels with respect to their susceptibility to developing cleavage fractures was effectively masked by numerous variables. It was therefore necessary to investigate the effect of some of these variables, since no quantitative data on them were available.

In this section are reported the effect of the following variables on the development of brittle or cleavage fractures, determined by the cleavage-tear tests:

- Eccentricity of loading.
- Depth and thickness of section.
- Depth of notch.
- Angle of notch.
- Radius at base of notch.
- Heat-treatments.
- Rate of load application.
- Temperature of testing.

Unless otherwise stated, the following test conditions were used: The steels were tested in full thickness, in the as-received condition and in a direction so that the fractures produced were transverse to the direction of rolling of the plate; the depth of notch was 3/16 in. and formed by cold-pressing; the angle of notch was 45 deg, with 0.0015 in. radius at the base of the notch; the depth of the test coupon was 6 in., and its thickness 1 in.; the nominal eccentricity of loading was 4.5 in. (type D coupon, Fig. 1); the

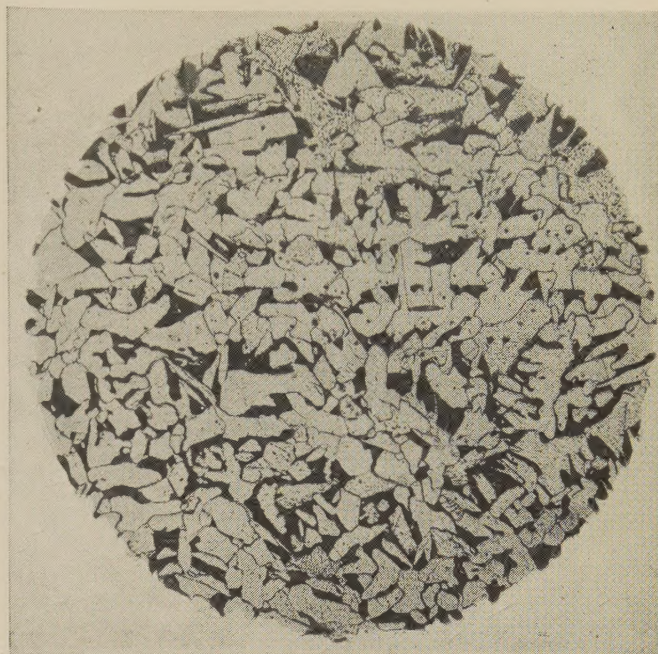


FIG. 5 (a) MICROSTRUCTURE OF STEEL 7 IN AS-RECEIVED CONDITION;
×100; PICRAL-NITAL ETCHANT

FIG. 5 (b) MICROSTRUCTURE OF STEEL 15 IN AS-RECEIVED CONDITION;
×100; PICRAL-NITAL ETCHANT

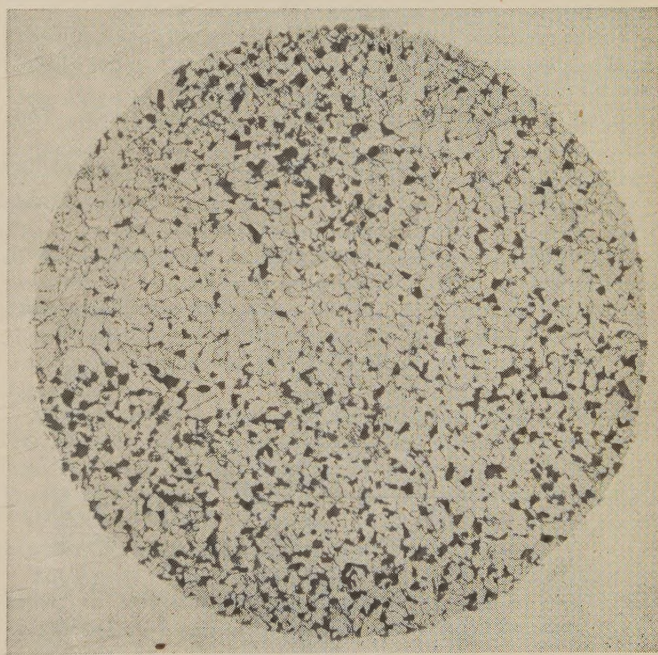


FIG. 5 (c) MICROSTRUCTURE OF STEEL 17 IN AS-RECEIVED CONDITION;
×100; PICRAL-NITAL ETCHANT

FIG. 5 (d) MICROSTRUCTURE OF STEEL 16 IN AS-RECEIVED CONDITION;
×100; PICRAL-NITAL ETCHANT

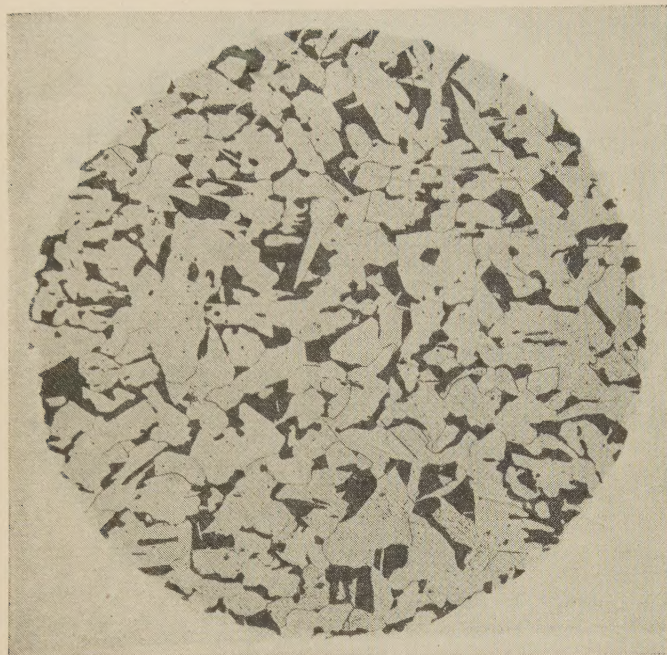


FIG. 5 (e) MICROSTRUCTURE OF STEEL 20 IN AS-RECEIVED CONDITION; $\times 100$; PICRAL-NITAL ETCHANT

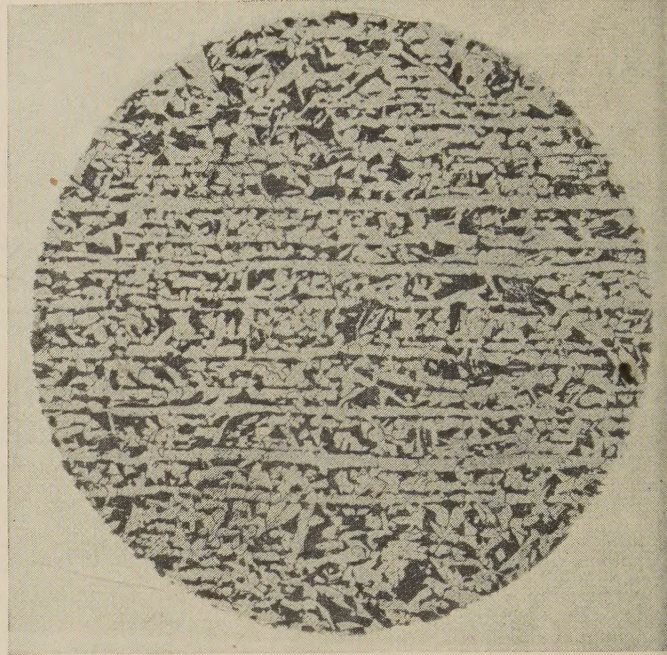


FIG. 5 (f) MICROSTRUCTURE OF STEEL 22 IN AS-RECEIVED CONDITION; $\times 100$; PICRAL-NITAL ETCHANT

temperature of testing was 32 F; and the rate of load application was 2 ipm of crosshead separation of the tensile machine.

Unless otherwise stated, the unit breaking load values, if listed in the tables as single values, are averages of two or more tests, and the fractures obtained are of the cleavage type.

Effect of Eccentricity of Loading on Breaking Loads. The

effect of eccentricity of loading on breaking loads was investigated by a series of tests on $\frac{1}{2}$ to $1\frac{1}{4}$ -in-thick plates, using the various types of test coupons shown in Fig. 1, for eccentricities of loading from 0 to 13 in., and 45 and 60-deg notches. Steels 12, 13, 15, and 17 were used for the coupons with 60-deg notch, and the breaking loads were determined for fractures in two directions: parallel

TABLE 4 EFFECT OF LOAD ECCENTRICITY, e , ON UNIT BREAKING LOAD, P/A

(A) NOTCH ANGLE, 60°

Eccentricity, e , in.*	Unit Breaking Loads, P/A , Psi				Computed, P/A
	Steel 12, $\frac{3}{4}$ in.	Steels 15 and 17, 1 in.	Steel 13, $1\frac{1}{4}$ in.	Average, P/A	
Fractures Transverse to Rolling Direction					
0	...	46,000	...	46,000	46,200
1	28,200	...	...	28,200	28,900
1.5	24,600	24,100	23,000	24,000	24,000
3	17,600	...	17,000	17,300	16,600
4.5	12,700	13,000	13,100	13,000	12,600
7†	9,300	...	8,700	9,000	8,900
9	7,300	...	7,500	7,400	7,300
13	6,600 (shear)	...	5,400	5,400	5,300
Fractures Parallel to Rolling Direction					
0	...	45,700	...	45,700	45,700
1	27,000	...	...	27,000	26,600
1.5	23,000	24,100	21,400	22,800	22,000
3	16,700	...	16,300	16,500	14,500
4.5	12,100	11,000	12,300	11,800	11,000
7†	8,000	...	8,000	8,000	7,600
9	...	...	6,300	6,300	6,100
13	4,600	...	4,400	4,500	4,400

(B) NOTCH ANGLE, 45°

Fractures Transverse to Rolling Direction				
	Steel 22, $\frac{1}{2}$ in.	Steel 17, 1 in.		
1.5	22,600-24,000	21,700-22,500	22,700	23,200
3	16,000-18,000	15,600-16,200	16,400	16,100
4.5	11,800-12,300	12,000-12,200	12,100	12,200
9	6,800-7,200	6,300-6,600	6,700	7,000

* The values of e listed above are nominal, to which $h/2$ or 0.094 in. should be added to obtain the actual eccentricity (Fig. 1).

† The coupon for these tests was the same as coupon Type F (Fig. 1) except its dimensions e and m were 1 in. longer than those of Type F coupon.

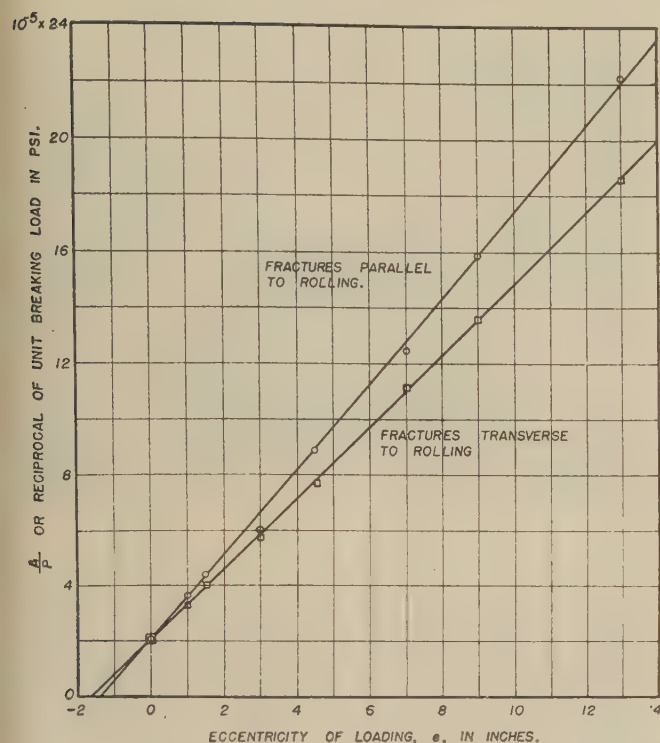


FIG. 6 EFFECT OF ECCENTRIC LOADING ON UNIT BREAKING LOADS

and transverse to the direction of rolling of the plate. For the 45-deg notch, steels 17 and 22 were used, with fractures transverse to direction of rolling of the plate. The results are given in Table 4.

Fig. 6 shows the reciprocal of unit breaking loads for transverse and parallel fractures, as a function of eccentricity of loading, for the coupons containing the 60-deg notch. The resulting curves are straight lines, indicating that P/A is inversely proportional to e . It is to be noted that the slope of the curve for transverse fractures is considerably different from that for the parallel fractures. This indicates that the anisotropic properties of rolled steel plates can be clearly revealed by the cleavage-tear tests, and that the resistance to brittle fractures in the direction parallel to rolling is considerably less than in the direction transverse to rolling of the plate, the anisotropy becoming more pronounced as the eccentricity of loading is increased. This is evident from Equations [7] and [8] which follow. Equation [8] covers fractures parallel to the rolling direction and contains the anisotropy factor q , the value of which for the steels tested is 0.752/0.617 or 1.23. In deriving this value, q is assumed to have an arbitrary value of 1 for fractures transverse to the direction of rolling. The test results appear to be more consistent for fractures in the transverse direction than for those in the direction parallel to rolling of the plate.

It was found that Equation [5] could be used for expressing the relationship between the load eccentricity and experimentally determined unit breaking loads, P/A . The numerical values of S , n , and q were found from the test data, and substituted in Equation [5] yielding the following equations

Transverse fractures

$$60\text{-deg notch: } \frac{P}{A} = \frac{49,200}{1 + 0.617 \left(\frac{6e}{d-h} \right)} \dots\dots [7]$$

Parallel fractures

$$60\text{-deg notch: } \frac{P}{A} = \frac{49,200}{1 + 0.752 \left(\frac{6e}{d-h} \right)} \dots\dots [8]$$

Transverse fractures

$$45\text{-deg notch: } \frac{P}{A} = \frac{48,000}{1 + 0.617 \left(\frac{6e}{d-h} \right)} \dots\dots [9]$$

The P/A values computed by these equations are also listed in Table 4. The equations show that the value of the factor n remains constant throughout the entire range of the tests. This indicates that no curved-beam conditions exist for the values of e , d , and h used, and that in so far as cleavage fractures are concerned the total deflections produced prior to fracturing are rather small. It is also obvious that as the load eccentricity is increased beyond certain limits, the effectiveness of the notch is reduced (see Equation [6]), and thereby the deflections become appreciable, causing shearing to occur. In the latter case, the factor v should be applied, as indicated by Equation [4]. Such a condition appears to have been reached in the tests where the values of $e/(d-h)$ exceeded about 2.3.

An examination of Equations [7], [8], and [9] reveals that the numerators in these equations are also constant throughout the entire range of the tests and could be considered to represent the cleavage strengths, in psi, of the steels tested in the presence of the corresponding notches, since P/A equals the numerator in each case if e is assumed to be zero. It appears that the numerical values of S in Equations [7], [8], and [9] could also be considered as representing the breaking strength of these steels in the absence of a notch, but under the same state of stress which would have prevailed had the notch been present.

Figs. 7 (a) to (d) show the appearance of cleavage-tear coupons, after fracturing at 72 F, for eccentricities of loading of 0, 3, 7, and 9 in., respectively. The coupons were coated with lime whitewash to show the stress pattern, and, in addition, reference lines were drawn on the coupon shown in Fig. 7 (b), for measuring the deformation produced by the fracture. It is to be noted from Fig. 7 (a), that even if the tensile pull is applied on the centroid of the coupon, some eccentricity of loading is induced by the presence of the notch on the edge of the coupon, as indicated by Equation [6]. A comparison of Figs. 7 (b), (c), and (d) shows that the neutral axis of the coupon moves closer to the centroid with increasing eccentricity of loading, as is indicated by Equation [6].

Effect of Section Depth and Thickness on Breaking Loads. The effect of coupon or section depth and thickness on the unit breaking loads was investigated by using coupons of various depths shown in Fig. 1, the range of coupon depths covered being 1 to 24 in. Steels 7, 15, 17, 20, and 22 were used for the tests representing a thickness range of $1/2$ to $2 1/4$ in., inclusive.

The results of these tests are listed in Table 5 and those obtained on 1-in.-thick coupons graphically presented in Fig. 8. It is to be noted that as the depth of the coupon decreases from the value of 6 in., the unit breaking load decreases, and also increasing the coupon depth beyond about 6 in. has practically no effect on the unit breaking load.

Equation [9] derived from loading-eccentricity tests described in the preceding section for 45-deg notch was found to be applicable, except it had to be modified to satisfy the condition that for coupons 6 in. or greater in depth the P/A value remains practically unchanged. The modified equation applies where the load-eccentricity values are such that the term $e - (d-h)/2$ is greater than -3 . If the latter is less than -3 the point of load application would be below the centroid of the 6-in. section, and

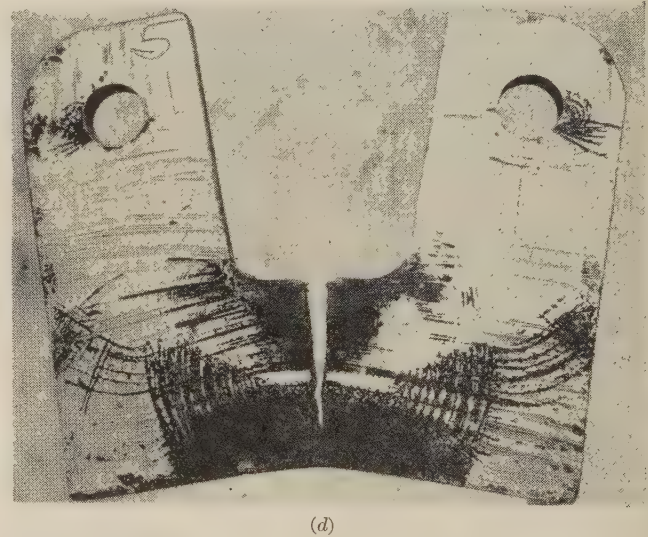
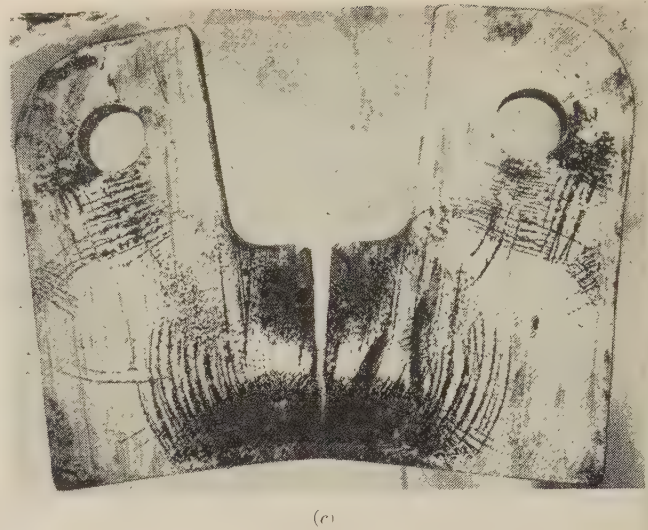
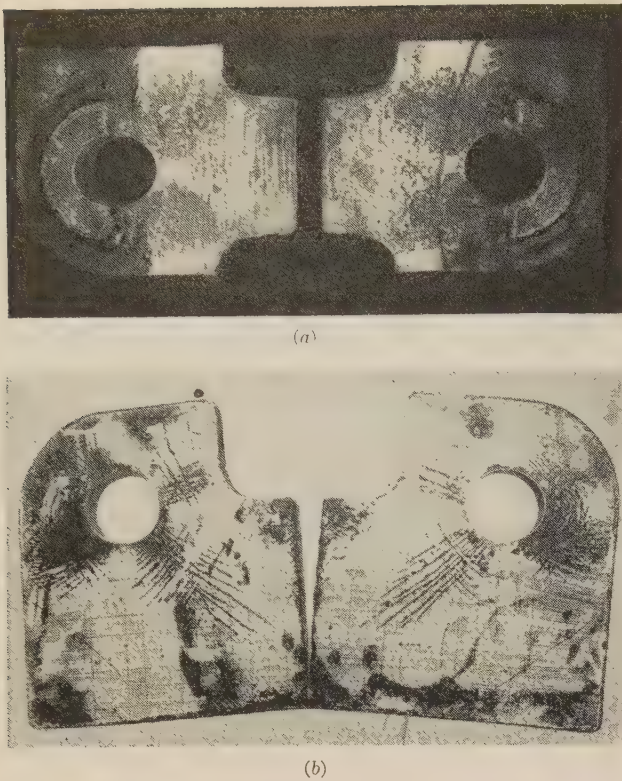


FIG. 7 CLEAVAGE-TEAR COUPONS AFTER FRACTURE, WITH LIMEWASH COATING APPLIED, TO SHOW STRESS PATTERN

(a, Coupon has a depth of 4 in. and centric loading; b, has a depth of 6 in. and an eccentricity of loading of 3 in. Notch angle was 60 deg in both coupons. c and d, Coupon depth was 6 in. and notch angle 60 deg in both cases, with eccentricity of loading of 7 and 9 in., respectively.)

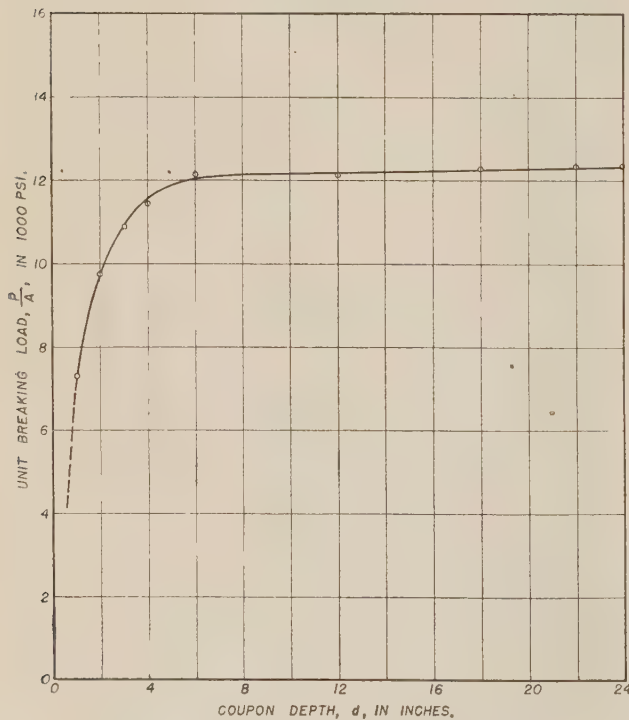


FIG. 8 EFFECT OF COUPON DEPTH ON UNIT BREAKING LOADS

it appears thereby the effect of the notch is nullified. The modified equation is

$$\frac{P}{A} = \frac{48,000}{1 + 0.617 \left[\left(\frac{6e}{d-h} \right) + k(d-h-6) \left(\frac{e}{d-h} - 0.50 \right) \right]} \quad [10]$$

where

$k = 0$, if net section depth, $d-h$, is less than 6 in.

$k = 1$, if net section depth, $d-h$, is 6 in. or greater, and $e - (d-h)/2$ is greater than -3 .

The P/A values computed by Equation [10] are listed in Table 5 and agree well with the experimentally determined values for the entire range of coupon depths and thicknesses covered. The tensile testing machine of 300,000 lb capacity, available for these tests, did not permit testing coupons in 1-in-thick plates greater than 24 in. in depth, with the load eccentricity used. It is believed that the results obtained on coupons of depths greater than the foregoing would conform to Equation [10].

It would appear from what has been said that a net section depth of about 6 in. is required to obtain the required rigidity

TABLE 5 EFFECT OF DEPTH AND THICKNESS OF COUPON, ON UNIT BREAKING LOAD, P/A

Coupon depth, d , in.	Unit Breaking Loads, P/A , Psi				
	Steel 22, $\frac{1}{2}$ In.	Steel 17, 1 In.	Steel 20, $2\frac{1}{4}$ In.	Average, P/A	Computed, P/A
$\frac{1}{16}$	5,000–5,200	7,200*	5,700–6,500	5,500	5,300
$\frac{2}{16}$		8,700–10,700		9,700	8,100
$\frac{3}{16}$	9,400–10,800	10,600–11,000	9,800–10,600	10,400	9,700
$\frac{4}{16}$		11,000–11,700		11,400	10,900
6	11,600–13,300	12,000–12,300	11,600–12,000	12,100	12,200
10			12,400–12,800	12,600	12,300
12	13,000–13,600	12,000–12,400		12,600	12,300
18	10,600–12,400	12,200–12,400		12,300	12,300
22		12,200–12,500		12,350	12,350
24		12,300–12,500		12,400	12,350

* Partly shear, due to excessive deflection.

TABLE 6 EFFECT OF NOTCH DEPTH h , ON UNIT BREAKING LOAD P/A , AND ON NOMINAL CLEAVAGE STRENGTH, S

Notch depth, h , in.	Experimental		Computed P/A , psi	Computed nominal cleavage strength, S , psi
	P/A range, psi	P/A average, psi		
$\frac{1}{32}$	22,500–23,000	22,750	23,000	90,500
$\frac{1}{16}$	14,300–14,700	14,500	14,500	57,300
$\frac{3}{32}$	12,800–12,800	12,800	13,000	50,400
$\frac{1}{8}$	12,000–12,600	12,300	12,300	48,300
$\frac{5}{32}$	12,000–12,100	12,050	12,150	48,000
$\frac{3}{16}$	12,000–12,300	12,150	12,150	48,000
$\frac{1}{4}$	12,000–12,100	12,050	12,000	48,000
1			10,200	48,000

and to compensate for the stress gradients created by the notch, and that for sections 6 in. or greater in depth, the P/A value becomes independent of the size of the section, within the limits investigated. It is apparent, however, that the foregoing need be somewhat modified in cases where the degree of triaxiality is appreciably increased by a pronounced increase in the ratio of the section thickness to section depth. The use of large test coupons has recently been advocated by some investigators (3) on the basis of other considerations. It would appear also that the numerator in Equation [10], which is 48,000 psi, is the nominal cleavage strength of the steels for the notch geometry used, since, for coupons 6 in. or greater in net depth, the value of P/A approaches that of the numerator as the term $e - (d - h)/2$ approaches the value of -3 . This appears to explain why the unit breaking load does not change materially as the thickness of coupon is varied or its net depth is increased beyond 6 in.

It follows from this that a notch of specified geometry and orientation establishes a state of stress which in turn establishes a well-defined load level to initiate cleavage fracturing. A small notch at the edge of a rectangular section appears to modify the breaking strength or the load-carrying capacity of the section, if the loading is such as to make the notch effective, regardless of the depth of the section.

Effect of Notch Depth on Breaking Loads. The effect of notch depth was determined by a series of tests, in which D-type coupons, shown in Fig. 1, made of steel 17 were used, except that the notch depths were varied from $\frac{1}{32}$ to $\frac{1}{4}$ in. The results of the tests are listed in Table 6, and graphically presented in Fig. 9. In Table 6 are also included the computed nominal cleavage strength, S , and P/A values for various notch depths.

The equation for the curve in Fig. 9 expresses the effect of notch depth variations. By modifying Equation [10] by the term just mentioned, the following equation is obtained

$$\frac{P}{A} = \frac{48,000 \left[1 + 0.71m \left(1.245 - \frac{h - 0.031}{h^{1.24}} \right) \right]}{1 + 0.617 \left[\left(\frac{6e}{d - h} \right) + k(d - h - 6) \left(\frac{e}{d - h} - 0.50 \right) \right]}$$

.....[11]

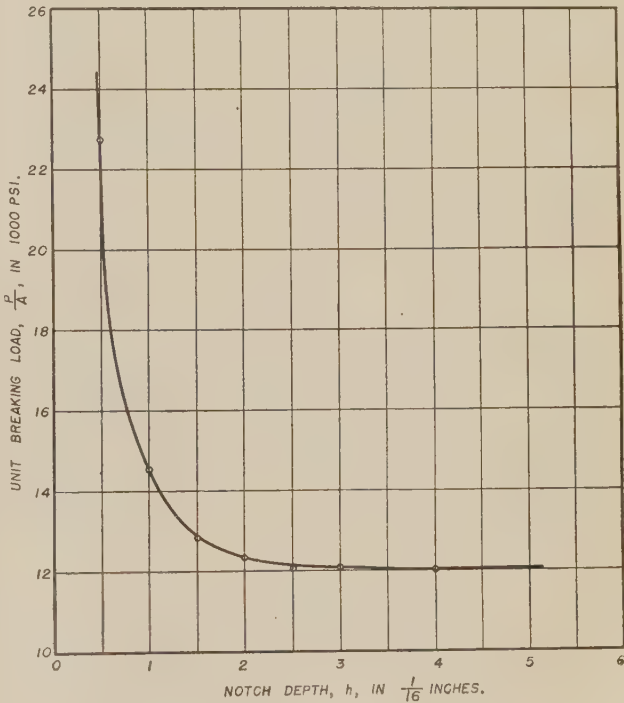


FIG. 9 EFFECT OF NOTCH DEPTH ON UNIT BREAKING LOADS

in which

- $m = 1$, if notch depth is less than $\frac{5}{32}$ in.
- $m = 0$, if notch depth is $\frac{5}{32}$ in. or greater

It is to be noted that in the preceding section the results were found to be consistent by using the same notch depth, while the coupon depths and thicknesses were varied through a fairly wide range. This indicates that the effect of the geometry of the notch on breaking loads is not directly related to the effects produced by the geometry of the test coupon. Equation [11] shows

that changing the notch depth changes the nominal stress by an amount proportional to the parabolic relation: $(h - b)/h^c$, in which b and c are constants. It is evident from the latter that P/A is practically unchanged if the value of h becomes large, and that it approaches infinity as the value of h approaches zero, indicating that no cleavage fractures develop in absence of a notch.

Table 6 shows that the nominal cleavage strengths increase appreciably as the notches become shallower, the value of S for a notch depth of $1/32$ in. being 90,500 psi. This would indicate that the extent of stress concentration decreases as the notch depth decreases. It is to be noted that cleavage strengths are independent of notch depths greater than about $5/32$ in., and that a notch as shallow as $1/32$ in. is sufficient to initiate a rupture or cleavage crack, although at much higher loads than the deeper notches. It follows then that the so-called "notch severity" is not as important a factor in large sections as has been suspected. On this basis, it is obvious that a sharp notch of $5/32$ in. or greater in depth is capable of concentrating stresses to such a degree as to reduce the nominal cleavage strength of steel below its shear strength, thereby favoring cleavage or brittle fractures to occur at relatively low loads, if the section is 6 in. or greater in depth and is of sufficient thickness to support development of cleavage cracks. The implications of this conclusion in design work are numerous, discussion of which is beyond the scope of this paper.

Effect of Notch Angle on Breaking Loads. For determining the effect of notch angle on breaking loads a series of tests were made using D-type coupons from steels 15 and 17, with angles of notch varying from 30 to 120 deg. Most of the tests were conducted at 32 F, and some at room temperature. The experimental data are summarized in Table 7 and graphically presented in Fig. 10.

The equation expressing P/A as a function of the tangent of one half the notch angle, $\tan N/2$, is as follows

$$\frac{P}{A} = 12,150 \left[1 + 0.29 \tan \frac{N}{2} \left(\tan \frac{N}{2} - \tan 22.5 \right) \right] \dots [12]$$

Equation [12] indicates that, as the notch angle is changed, the breaking load is modified by a component of the stress proportional to: $\tan N/2 [\tan N/2 - \tan 22.5]$. Although the $\tan N/2$ versus P/A curve is parabolic in type, the P/A value for notch angles 0-45 deg can be considered as being approximately constant and equal to 12,150 psi for the test coupon used.

By combining Equations [11] and [12] the following more general equation is obtained

$$\frac{P}{A} = \frac{48,000 \left[1 + 0.71m \left(1.245 \frac{h - 0.031}{h^{1.24}} \right) \right] \left(1 - 0.12 \tan \frac{N}{2} + 0.29 \tan^2 \frac{N}{2} \right)}{1 + 0.617 \left[\left(\frac{6e}{d-h} \right) + k(d-h-6) \left(\frac{e}{d-h} - 0.50 \right) \right]} \dots [13]$$

TABLE 7 EFFECT OF ANGLE OF NOTCH N , ON UNIT BREAKING LOAD, P/A , AND ON RATIO OF YIELD TO BREAKING LOADS

Notch angle, N , °	$\tan N/2$	Experimental		Computed, P/A , psi	Ratio of yield load to breaking load	Fracture
		P/A Range, psi	P/A Average, psi			
0	0.000	12,000-12,200	12,100	12,150		
30	0.268	12,000-12,300	12,150	12,100	0.94	Cleavage
45	0.414	12,300-12,700	12,500	12,150	0.93	Cleavage
60	0.578	13,000-13,400	13,200	12,600	0.90	Cleavage
75	0.793	13,800-14,600	14,200	13,100	0.86	Cleavage
90	1.000	14,800-15,400	15,100	14,200	0.80	Cleavage
98	1.150	15,500-16,900	16,200	15,100	0.75	Shear-cleavage
105	1.303	19,000-20,600	19,800	16,300	0.71	Shear
120	1.732			20,200	0.56	Shear

It is of interest to note that in Equations [7] and [9], which cover load-eccentricity tests with 60 and 45-deg notches, the value of the factor n remains constant, although the nominal cleavage strengths of the steels are different for the two notches. This indicates that the curves A/P versus e for cleavage fractures obtained with notches of different angularity are parallel to each other, and that only the numerator of the equation need be changed if the notch angle is changed. This would be predicted by Equation [13]. On this basis, Equation [7] was computed from Equation [9] by the use of the notch-angularity term given, which computation yielded the value of about 49,300 psi for the numerator of Equation [7] as compared with the experimentally determined value of 49,200 psi. The agreement is not bad.

The unit breaking loads computed in accordance with Equation [13] are included in Table 7, and are in close agreement with the experimental values. Equation [4] need not then be applied for computing the shear-fracture loads, since in this case the slope of the $\tan N/2$ versus P/A curve applies the necessary correction.

The experimental data show that with angles of notch of about 90 deg or less, the initial fractures are mostly of the brittle or cleavage type, whereas those with notch angles greater than 90 deg are of the shear or the ductile type. A straight notch of a width of about 0.035 in. made by the cut of a regular hacksaw was found to require about the same unit breaking load as a notch of 90 deg. In this connection it is of importance to note that fillets 90 deg or less in angularity, with small radii at the base, are capable of starting cleavage or fatigue cracks. The effect of notch radius on breaking loads is discussed in detail in the next section.

In conjunction with determination of breaking loads for notches of different angularity, the yield loads were also determined in each case. In these tests yielding occurs progressively, it first appearing at the notch. The yield load just referred to is the load at which the first evidence of yielding is observed by flaking off of the scale at or near the notch. It was found that the yield load was approximately the same for all notches, ranging from 10,800 to 11,800 psi, with 11,300 psi as the average. The ratio of the yield load to the breaking load for each notch angle is included in Table 7, and is also shown graphically in Fig. 10.

An examination of values of this ratio shows that where the ratio of the yield load to breaking load is about 0.80 or higher, cleavage fractures occur, and that where this ratio is lower than 0.80, shear fractures predominate. It is also apparent, that as the notch angle approaches zero, the yield load almost coincides with the breaking load. It follows then that, other conditions being the same, cracks if subjected to stresses of sufficient intensity are likely to extend themselves in the form of cleavage fractures. It is also apparent that a finite unit breaking load is required to pro-

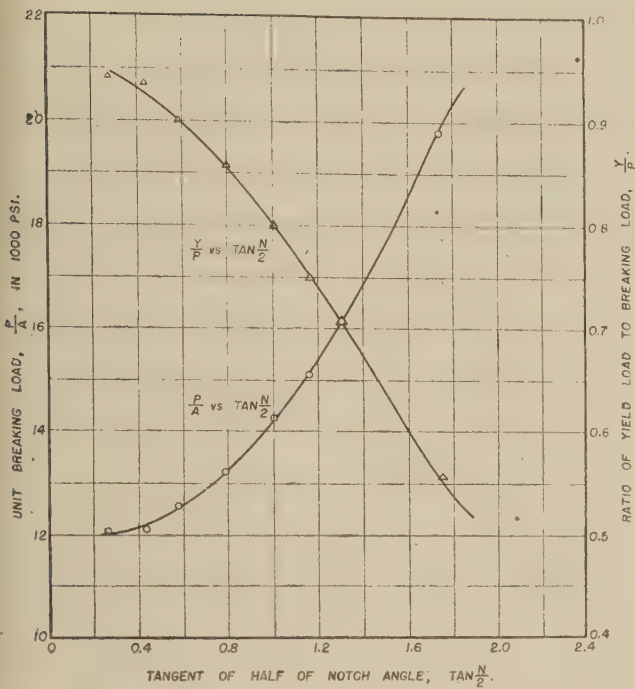


FIG. 10 EFFECT OF NOTCH ANGLE ON UNIT BREAKING LOADS AND ON RATIO OF YIELD TO BREAKING LOADS

duce fractures even if the angle of notch approaches zero. Zero angle of notch can be assumed to be approached by cracks, on which basis the P/A value computed by Equation [13] for $N = 0$ is 48,000 psi.

In light of the foregoing data, let us consider the maximum deflection within the elastic range obtainable at the center of a simple beam, which is supported at the ends and loaded in the center with a single load. This deflection can be computed by the following flexure formula

$$F = \frac{Pl^3}{48EI}$$

where

- F = maximum deflection at center of beam, in.
- P = applied load, lb
- l = length of beam between supports, in.
- E = Young's modulus of elasticity = 30×10^6 psi
- I = moment of inertia, in.⁴

The coupon used in the tests summarized in Table 1 was a beam of this type, and was 1 in. thick, $3\frac{1}{2}$ in. deep, and had a span of 16 in. The data presented in Table 7 indicate that in sharp notches the yield point and the breaking load practically coin-

cide with each other. Assuming, then, P to have the value of about 30,000 lb, which load induces a tensile stress of about 60,000 psi at the notched edge of the beam, a deflection of about 0.025 in. is possible within the elastic range, according to the equation cited, assuming that the breaking strength and the yield point (proportional limit) are brought together by the notch. The curve F versus R in Fig. 4 shows that this amount of deflection would be obtained if the radius at the base of the notch was about 0.001 in., from which it follows that with the latter radius at the base of the notch the beam would break but still would be within the elastic range. In a subsequent section of the paper it is shown that the radius at the base of cleavage fractures is of the order of three millionths of an inch. By extrapolation, it is found from Fig. 4 that the deflection corresponding to this notch radius is about 0.0003 in. It is obvious therefore that no appreciable yielding is possible in fractures of strictly cleavage type, since the radius at the base of such cracks is much smaller than the given value, and that under the circumstances such cracks extend themselves with cleavage fractures. The latter confirms conclusions reached by notch-angularity data, just discussed.

Effect of Notch Radius on Breaking Loads. The effect of the radius at the base of the notch on breaking loads was studied through a series of tests on steel 17, using D-type coupons, in which notch radii varied from 0.0015 to 0.10 in. In Table 8 are listed the experimental and computed unit breaking loads for each notch. Fig. 11 is a log-log plot, showing P/A as a function of the notch radius R . The resultant curve is a straight line, its equation being

$$P/A = 25,300 R^{0.118} \dots [14]$$

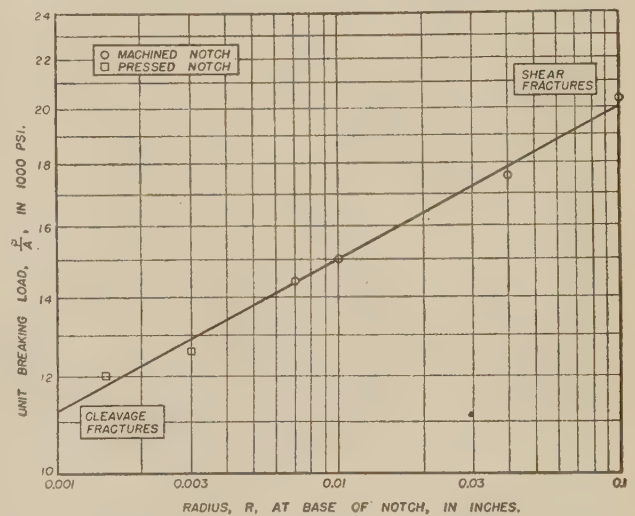


FIG. 11 EFFECT OF NOTCH RADIUS ON UNIT BREAKING LOADS

TABLE 8 EFFECT OF NOTCH RADIUS R , ON UNIT BREAKING LOAD P/A (TEMPERATURE OF TEST: 72 F)

Notch Type	Notch radius, R , in.	Experimental		Computed P/A , psi.	Fracture*
		P/A Range, psi	P/A Average, psi		
Base of a cleavage crack	3×10^{-6}		...	6,020	(B)
	1×10^{-4}		...	8,950	(B)
	1×10^{-3}		...	11,600	(B)
	1.5×10^{-3}	11,700-12,200	11,950	12,150	B
Pressed notch, 45°	3×10^{-3}	12,600-12,600	12,600	13,200	B
Pressed notch, 45°	7×10^{-3}	13,400-15,100	14,250	14,500	B
Machined V notch, 45°	1×10^{-2}	14,500-15,500	15,000	15,100	S, B
Machined Izod V, 45°	4×10^{-2}	16,850-18,150	17,500	17,650	S, B
Machined Charpy keyhole	1×10^{-1}	19,000-21,500	20,300	19,500	S
Drilled round hole					

* S designates shear and B brittle or cleavage fractures. Letters in parentheses designate probable type of fracture.

It is to be noted that Equation [14] is identical in form to theoretical equations for expressing natural strain in terms of the tensile stress (4). The constant 25,300 in Equation [14] can be written in the following form: $25,300 = 12,150/(0.0015)^{0.113}$, where 12,150 is the experimentally determined unit breaking load for a notch with 0.0015 in. root radius. This expression $1/R^{0.113}$ can be considered as being an index of stress concentration at the notch as approximate stress measurements have shown. It appears that R is independent of the notch angle, as is indicated by the data given in Tables 1 and 8, and of coupon depth as indicated by the data in Table 5. The value of $1/(0.0015)^{0.113}$ is 2.085.

Insertion of these values in Equation [13] yields the following equation³

$$\frac{P}{A} = \frac{48,000 \left[1 + 0.71m \left(1.245 - \frac{h - 0.031}{h^{1.24}} \right) \right] \left(1 - 0.12 \tan \frac{N}{2} + 0.29 \tan^2 \frac{N}{2} \right) (2.085 R^{0.113})}{1 + 0.617 \left[\left(\frac{6e}{d-h} \right) + k(d-h-6) \left(\frac{e}{d-h} - 0.50 \right) \right]} \quad [15]$$

It is apparent from Equations [14] and [15] that P/A becomes zero if R is zero. However, since finite loads are required to continue extension of cracks into sound sections, it is obvious that the radius at the base of a crack has a finite value. Numerous cleavage cracks were examined at high magnifications with the object of measuring the radius at the base of such cracks. It was found that the radii varied considerably, the smallest being of the order of 3×10^{-6} in. The breaking strength of the steel section containing a $3/16$ -in.-long cleavage crack computed by Equation [15] amounts to 23,700 psi. The order of the stress at the base of such a crack is obtained by multiplying the constant 48,000 in Equation [15] by 4.21, which is the reciprocal of $R^{0.113}$, with $R = 3 \times 10^{-6}$. According to the foregoing, this stress is in excess of about 200,000 psi, indicating that the values of technical cohesive strength of the steel are reached at the base of cleavage cracks. The relative stress-concentration ratio on this basis would then be greater than 9. In the foregoing it is assumed that the steel is at a temperature below its transition-temperature range.

Fig. 11 shows that the unit breaking loads for small root radii of notches made by cold-pressing and those for notches with large root radii prepared by machining or drilling conform to the straight-line relationship of P/A versus R , within the experimental accuracy. This indicates that the breaking loads are not affected materially by the shallow cold-working found at the base of the pressed notches. Undoubtedly, some localized cold-working occurs at the base of all notches before fracturing occurs, so that the small amount of cold-working present in the pressed notches prior to stressing should not change the situation materially. This is also indicated by Fig. 4 and the data given in Table 1.

In this connection, the test data also show that notch radii smaller than the radius of the conventional Izod V-notch (0.01 in.) favor development of cleavage or brittle fractures, and that notches with Izod or Charpy radii produce a combination of cleavage and shear fractures. Notches with radii greater than the latter notches favor shear fractures. This is illustrated in Figs. 12(a) and (b). The shear fracture in Fig. 12(a) started from a notch which had a radius of 0.10 in. at its base, whereas the cleavage-shear fracture in Fig. 12(b) started from a notch with 0.04 in. radius at its base.

The nominal breaking strength of 23,700 psi referred to is only about 40 per cent of the axial tensile strength of the steels tested, determined by the conventional method. It follows then that cracks, if present, are very dangerous since relatively a much smaller load will be required to extend them in the sound struc-

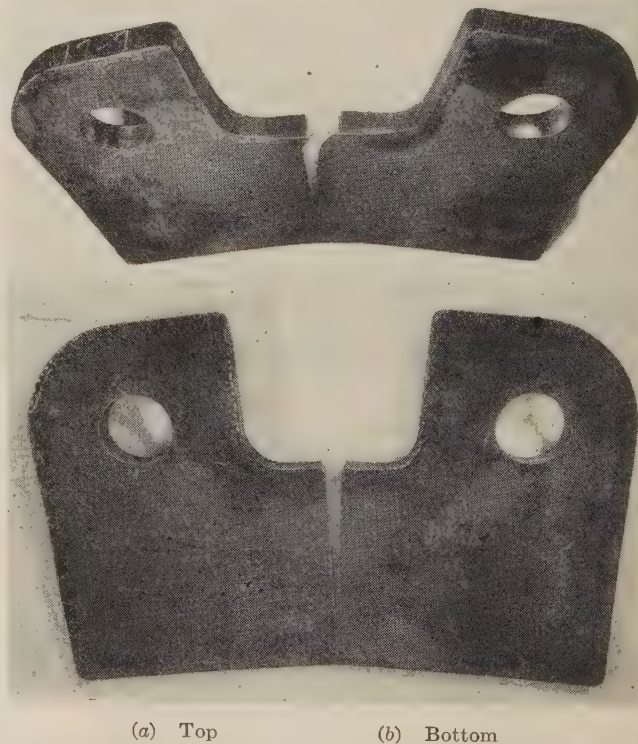
ture, and because the breaking load needed progressively decreases as the depth of the crack increases, due to resultant increase in eccentricity of loading.

Sharp notches, such as surface fissures in forgings, castings, or rolled plates, also weld underbead cracks, weld undercuts, and similar surface irregularities fall under this category. It is obvious that they should be avoided in all cases possible, since cleavage fractures are likely to originate from them at relatively low load levels because of the smallness of the radius at the base of these irregularities. Incipient fatigue cracks can be considered as being small cleavage fractures.

Effect of Heat-Treatment on Breaking Loads. In Table 9 are summarized the unit breaking loads obtained on steel 17 after

various heat-treatments, including annealing, normalizing, water-quenching, and stress-relieving, and also on manual and submerged arc-weld seams in the as-welded and stress-relieved conditions. Notch angles of 45, 60, 90, and 120 deg were used to determine the effect of notch angularity on the behavior of the steel in various heat-treated conditions. For the weld seams the 45-deg notch was used.

In these tests the coupons were notched after heat-treatment. Of the heat-treatments used, normalizing consisted of heating to 1650 F for 1 hr and cooling in still air. Annealing was done by heating for 1 hr at 1650 F and furnace-cooling. Water-



(a) Top (b) Bottom

FIG. 12 TYPES OF SHEAR FRACTURES

(The shear fracture, shown in a, started from a 45-deg notch whose base radius was 0.10 in. in a coupon 6 in. deep and 1 in. thick when a load of 121,800 lb was applied with an eccentricity of loading of $4\frac{1}{2}$ in. $P/A = 20,300$ psi. Fracture: Shear. The fracture shown in b started from a 45-deg notch whose base radius was 0.04 in. in a coupon 6 in. deep and 1 in. thick when a load of 101,000 lb was applied with an eccentricity of loading of $4\frac{1}{2}$ in. $P/A = 16,850$ psi. Fracture: A combination of cleavage and shear. Both coupons were broken at 75 F.)

³ k and m have the same values in Equation [15] as those designated in previous sections.

TABLE 9 EFFECT OF HEAT-TREATMENT ON BREAKING LOADS

(a) Steel 17, in Conditions Indicated

Condition	P/A Range, psi	Average P/A , psi	$\frac{(P/A)_{HT}^*}{(P/A)_{AR}^*}$	Fracture
Notch angle, 45°				
As-received	11,800–13,100	12,150	1.00	Cleavage
Stress-relieved				
At 300° F.	11,400–12,200	11,800	0.97	Cleavage
At 1200° F.	11,000–11,250	11,100	0.92	Cleavage
Normalized	12,200–13,800	13,300	1.10	Cleavage-shear
Water quenched	16,300–17,500	16,900	1.40	Shear
Water quenched and stress-relieved at 1200° F.	16,000–17,300	16,650	1.37	Shear
Notch angle, 60°				
As-received	10,800–13,200	12,500	1.00	Cleavage
Normalized	15,000–18,900	17,600	1.41	Shear
Water quenched	17,000–24,000	20,700	1.66	Shear
Annealed	10,200–16,700	13,500	0.85–1.08	Cleavage-shear
Notch angle, 90°				
As-received	13,000–14,800	14,200	1.0	Cleavage
Normalized	13,600–14,700	14,020	1.0	Cleavage
Notch Angle, 120°				
As-received	18,100–21,000	19,800	1.0	Shear
Normalized	19,000–19,600	19,300	1.0	Shear

(b) Welded Joints, in Conditions Indicated

Notch angle, 45°				
Manual Welds				
As-welded	13,000	13,300	1.00	Cleavage
Stress-relieved				
At 300° F.	16,800	16,800	1.26	Cleavage-shear
At 1200° F.	17,300–17,800	17,600	1.32	Shear
Submerged-arc welds				
As-welded	12,500	12,500	1.00	Cleavage
Stress-relieved				
At 300° F.	12,000–12,500	12,250	1.00	Cleavage
At 1200° F.	14,850–16,600	15,750	1.26	Cleavage-shear

* $(P/A)_{HT}$ designates unit breaking load in heat-treated condition. $(P/A)_{AR}$ designates unit breaking load in as-rolled condition.

quenching consisted of heating for 1 hr at 1650 F, followed by quenching in water. The stress-relieving treatment consisted of heating for 1 hr at the temperatures indicated in Table 9.

It is to be noted from the test data given in Table 9 that normalizing and, especially, water-quenching materially increase the breaking loads and at the same time reduce markedly the tendency toward development of brittle fractures if the angle of notch is 60 deg or less. The effect of these heat-treatments is not detectable with angles of notch greater than about 60 deg. It was found that annealing produces some very erratic results, evidently due to the fact that this heat-treatment promotes segregation and ferrite banding in some of the steels, and thereby reduces their resistance to cleavage fracturing. Some further study in this direction is contemplated. Somewhat analogous results were also obtained in steel plates stress-relieved at 1200 F. In the case of welds, stress-relieving at 1200 F was found to be beneficial both to manual and to submerged arc welds. The latter heat-treatment increased the breaking loads as well as the resistance of both welds to propagation of brittle fractures. Heating for 1 hr at 300 F produced practically no effect on the plate materials but appeared to increase somewhat the resistance of the manual weld to cleavage-fracturing. The effect of heat-treatments on transition temperatures is discussed in a subsequent section of the paper.

The effect of heat-treatments is somewhat quantitatively expressed in terms of the ratio of the unit breaking load of the heat-treated pieces to that of the pieces in the as-received condition. These ratios are tabulated in the next to last column of Table 9, for comparison. It can be seen, for example, that with a notch angle of 60 deg the load required to produce cleavage fractures in steel 17 is increased 40 per cent by normalizing and 66 per cent by water-quenching treatments.

The effect of heat-treatment on the fractures produced in steel 17 is shown in Fig. 13. The cleavage fracture obtained in an 18-in-deep coupon in the as-received condition is shown in view 1. The cleavage-shear fracture seen in view 2 is in the normalized coupon of 12 in. depth and the predominantly shear fracture of conchoidal type shown in view 3 is in the water-quenched coupon of 12 in. depth. All coupons were 1 in. thick, had a 45-deg notch and were tested at 32 F.

The fractures obtained in welds are shown in Fig. 14. The view at the left is the fracture in the submerged-arc-weld seam, and that in the center is in the seam of a manual weld, both welds being in the as-welded conditions. The fracture in both of these cases is predominantly of cleavage type. The shear fracture obtained in the manual-weld seam, after stress-relieving for 1 hr at 1200 F, is at the extreme right. In the latter three coupons the

TABLE 10 EFFECT OF LOADING RATE ON BREAKING LOADS

(Fractures parallel to rolling direction. Test temperature: 75 F. Notch angle 60 deg. Notch radius, not determined)

Speed, ipm	P/A in psi	
	Steel 7	Steel 15
(A) Eccentricity of loading, 4.5 in.		
2.1		14000
1.8	13300	14100
1.5	15700	14500
1.2	13300	14900
0.7	15800	
0.25	13400	14900
0.07	15500	13400
Average P/A	14500	14200
(B) Eccentricity of loading, 3 in.		
2.2	...	20700
1.8	...	19500
1.4	...	21100
0.5	22500	
0.25	19600	18000
0.06	19100	19000
Average P/A	20400	19600



FIG. 14 APPEARANCE OF FRACTURES IN WELD SEAMS

Fracture obtained in submerged-arc weld seam, in as-welded condition is at extreme left. Fracture in center is in manual weld broken in as-welded condition. Fracture in these two coupons is predominantly cleavage type. Shear fracture seen at right, which runs at about 45-deg with vertical plane of coupon, is in manual-weld seam, stress-relieved for 1 hr at 1200 F. All coupons are 1 in. thick, 6 in. deep, had a 45-deg notch, an eccentricity of loading of $4/32$ in., and were broken at 32 F.

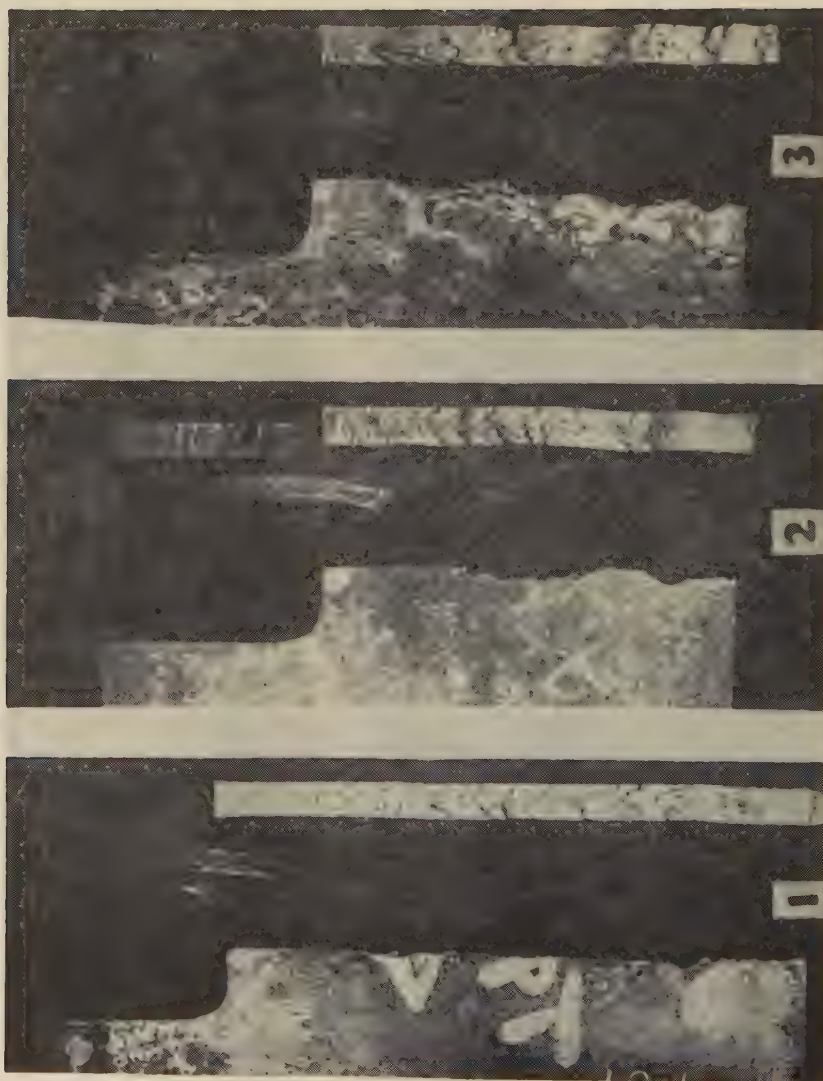


FIG. 13 TYPES OF FRACTURES

(Cleavage fracture in view 1 was obtained in as-rolled steel, coupon depth being 18 in. Cleavage-shear fracture in view 2 is in same steel in normalized condition, coupon depth being 12 in. Shear fracture of conchoidal type produced in same steel after water-quenching heat-treatment is shown in view 3. All coupons are 1 in. thick, had a 45-deg notch, radius at base, 0.0015 in., and were broken at 32 F.)

notch was 45 deg, the weld seam thickness 1 in., its depth 6 in., and 32 F as the testing temperature.

Effect of Loading Rate on Breaking Loads. The rate of load application was found to have no effect on the breaking loads, within the limits of loading rates investigated. The test results substantiating this are summarized in Table 10. Steels 7 and 15 were used in these tests, the coupon depth being 6 in. with eccentricities of loading of 3 in. and 4½ in. The loading rates used varied from 0.06 ipm to 2.2 ipm, the latter being the maximum speed of the crosshead separation of the tensile testing machine. In these tests a 60-deg notch, 3/16 in. deep, was used, and the testing temperature was about 75 F.

These tests were made in the early stages of this investigation, in which the coupons happened to be so oriented that the fractures fell parallel to the direction of rolling of the plates. Due to the anisotropy of these steel plates, discussed in a previous section of this paper, the unit breaking loads show considerable variations. Therefore these variations are believed to be the result of the directional properties of the steels, rather than the result of variations in the speed of load application.

It is known that the rate of load application has an important effect on breaking loads and on the resultant deformation in cases where the stresses are unidirectional, or where an appreciable deformation through yielding occurs. In cleavage-tear tests, the stresses are multiaxial and no appreciable yielding or deformation occurs if the fractures are of cleavage type. Consequently the cleavage fractures encountered in this investigation can be considered to be practically unaffected by the loading rates used.

Effect of Testing Temperature on Breaking Loads. A series of tests were made on 1-in-thick steels 7, 15, 16, and 17, using the type of test coupons, notching, and conditions listed in Table 11. The heat-treatments used were the same as previously described. The coupons were broken within the temperature range of -25 to 200 F with the object of determining the effect of temperature of testing on breaking loads. It was found that above certain testing temperatures within this range, the fractures obtained were of the shear type, whereas those obtained below these temperatures were of the cleavage type and required less load to cause fracturing. The temperatures at which these changes occurred were recognized as being the "transition temperatures" of

TABLE 11 EFFECT OF TESTING TEMPERATURE ON UNIT BREAKING LOADS AND FRACTURES (SINGLE TESTS)

Condition	Coupon depth, d, in.	Test temp deg F	P/A, psi	Fracture
Steel 15—Notch Angle, 45°				
As-received	6	125	16,700	Shear
As-received	6	110	16,300	Shear*
As-received	6	100	14,300	Cleavage-shear
As-received	6	95	12,300	Cleavage
As-received	6	32	12,100	Cleavage
Steel 15—Notch Angle, 60°				
As-received	6	200	16,300	Shear
As-received	6	110	17,000	Shear
As-received	6	105	15,900	Cleavage-shear
As-received	6	95	12,500	Cleavage
As-received	6	70	12,500	Cleavage
As-received	6	32	12,400	Cleavage
As-received	6	10	12,500	Cleavage
Normalized	6	50	18,500	Shear
Normalized	6	32	15,400	Cleavage
Normalized	6	10	14,400	Cleavage
Water quenched	6	35	19,200	Shear
Water quenched	6	0	17,000	Cleavage-shear
Steel 16—Notch Angle, 45°				
As-received	6	90	18,000	Double-shear
As-received	6	75	17,500	Cleavage-shear
As-received	6	70	17,000	Cleavage-shear
As-received	6	62	15,200	Cleavage†
As-received	6	32	14,700	Cleavage
As-received	6	0	14,200	Cleavage
As-received	6	-25	14,200	Cleavage
As-received	12	0	15,200	Cleavage
Normalized	12	0 and 32	20,000-21,000	Shear
Water quenched	12	32	25,000	Shear
Water quenched	12	0	17,800	Cleavage-shear
Steel 17—Notch Angle, 45°				
As-received	6	100	16,000	Shear
As-received	6	87	15,700	Shear*
As-received	6	79	12,300	Cleavage-shear
As-received	6	74	12,100	Cleavage
As-received	18	32	12,200	Cleavage
Normalized	6	75	17,600	Shear
Normalized	6	32	13,200	Cleavage
Normalized	18	32	13,800	Cleavage
Steel 7—Notch Angle, 45°				
As-received	6	130	17,800	Shear
As-received	6	125	17,200	Shear
As-received	6	125	17,300	Shear
As-received	6	115	17,500	Shear*
As-received	6	106	15,400	Cleavage†
As-received	6	90	13,800	Cleavage
As-received	6	32	13,000	Cleavage

* Cleavage fracture of about 1 in. length at bottom, where the section was subjected to compression due to bending.

† Partly shear fracture 1-2 in. in length at bottom, where the section was subjected to compression during fracturing, due to bending.

TABLE 12 CHARPY IMPACT TESTS AT VARIOUS TEMPERATURES (STANDARD KEYHOLE NOTCH COUPONS)

Test temp., ° F.	Steel 7				Steel 15			Steel 16			Steel 17			
	As-recd.†	As-recd.	Ann.	Norm.	WQ	As-recd.	Norm.	WQ	As-recd.	Ann.	Norm.	WQ	As-recd.	Ann.
-75	...	...	...	...	8, 10	...	...	...	...	...	...	...	...	...
-65	...	...	...	...	35, 3, 37	...	...	...	...	...	...	...	...	...
-50	...	...	...	...	34, 39	...	...	...	...	...	...	...	...	...
-35	...	...	...	...	41, 38	...	...	5, 7	25, 26	...	3, 4	24, 25	...	...
-20	...	...	...	...	39, 35	...	...	23, 25	...	12, 8	...	4, 3	...	...
-10	...	...	...	...	...	...	...	...	...	18, 22	...	15, 18, 12	36, 37	...
0	2, 2	...	...	...	...	...	...	...	...	...	...	...	...	...
15	13, 4, 5	5, 6	...	7, 8	44, 50	28, 29	...	9, 18	...	...	...	...	...	...
25	10, 11, 17	13, 6, 11	...	30, 29	...	...	...	...	26, 23	4, 5	22, 22	40, 39	...	...
32	18, 19, 7	21, 21	4, 4	...	...	...	...	...	...	7, 6	...	...	...	...
45	18, 22	18, 30	5, 20	37, 33	43, 49	...	...	24, 28	28, 26	20, 16	26, 22	38, 41	...	...
85	23, 21, 27	28, 30	11, 17	...	...	...	...	33, 37	35, 31	29, 23	35, 33	45, 41	...	...
Transition Temperature Range, ° F.	0 to 45	10 to 30	25 to 45	0 to 10	-80 to -65	-35 to -20	-85 to -75	-35 to 0	-20 to -10	25 to 32	-20 to -10	-65 to -50	...	...

*Impact values are in ft.-lb.

† As-Recd. = As-Received; Ann. = Annealed; Norm. = Normalized; WQ = Water Quenched.

TABLE 13 COMPARISON OF TRANSITION TEMPERATURES OF STEELS 7, 15, AND 17, AS DETERMINED BY CHARPY IMPACT AND CLEAVAGE-TEAR TESTS

	Transition temperatures, deg F			
	Cleavage-tear tests		Charpy impact tests	Transition temp. difference, deg F
	45-deg notch	60-deg notch	keyhole notch	
Steel 7				
As-received.....	100-115	...	0 to 45	90
Steel 15				
As-received.....	95-110	95-110	10 to 30	85
Annealed.....	...	...	25 to 45	...
Normalized.....	...	32-50	0 to 10	40
Water-quenched..	...	Below 0	-80 to -65	70
Steel 16				
As-received.....	65-80	...	-35 to -20	100
Normalized.....	Below 0	...	-85 to -75	...
Water-quenched..	Below 0	...	-35 to 0	...
Steel 17				
As-received.....	79-87	...	-20 to -10	100
Annealed.....	...	...	25 to 32	...
Normalized.....	32-75	...	-20 to -10	60
Water-quenched..	...	...	-65 to -50	...

the steels. In impact tests, the transition temperature is defined as being the temperature at which a marked change occurs in the amount of energy absorbed by fracturing, and at which the appearance of the fracture changes from the ductile to the brittle type (5). Therefore, Charpy impact tests were also made using the standard keyhole coupons, to establish transition temperatures of the steels by the latter tests. The fractures in both tests were transverse to the direction of rolling of the plate.

The results of the tests are summarized in Table 11, and those by the Charpy impact tests in Table 12. A comparison of the transition temperatures determined by the cleavage-tear and Charpy tests is shown in Table 13. Briefly, the transition-temperature ranges determined by the cleavage-tear tests for the steels in the as-received condition are as follows: For the rimmed steel, steel 7, 100-115 F; for the semi-killed steels 15 and 17, 95-110 F and 79-87 F, respectively; and for the titanium-deoxidized steel 16, 65-80 F.

It is to be noted from the tables that for the steels in the as-received condition there is a difference of 85-100 deg F between the transition temperatures determined by the Charpy and cleavage-tear tests, the latter showing consistently higher transition temperatures for all the steels tested. The difference between the transition temperatures for the steels in heat-treated conditions range from 40 to 75 deg F. Variations in notch angularity and coupon depths were found to have no appreciable effect, within the limits investigated.

Although the transition temperature range, as indicated by the impact tests, is somewhat indefinite for steel 7, a well-defined transition temperature range was revealed for this steel by the cleavage-tear tests, as the curves for these two types of test

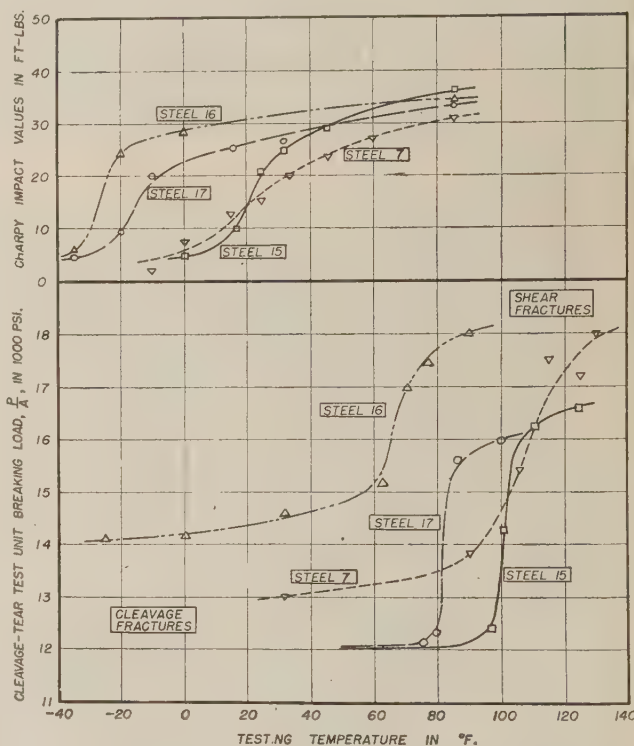


FIG. 15 EFFECT OF TESTING TEMPERATURE ON UNIT BREAKING LOADS AND DETERMINATION OF TRANSITION TEMPERATURES BY CHARPY IMPACT AND CLEAVAGE-TEAR TESTS

show in Fig. 15. Fig. 15(a) depicts the fractures obtained in D-type coupons of this steel, in the as-rolled condition, just below and above its transition temperature. The pair at the left was obtained by breaking the coupon at 106 F, and that at the right by testing at 115 F. The former shows a cleavage type of fracture and the latter the shear type. The transition temperature of this steel is therefore approximately within the temperature range of 100-115 F, as determined by the test conditions used. In Fig. 15(b) are shown additional fractures in the same steel at temperatures above and below the transition range, the pair at the left being obtained at 32 F, that in the center at 90 F, and the pair at the extreme right at 130 F. It is to be noted that the fractures below the transition range are of cleavage type, whereas those above that range are of shear type. Some scatter of results is possible near the transition temperatures, but the general trend is

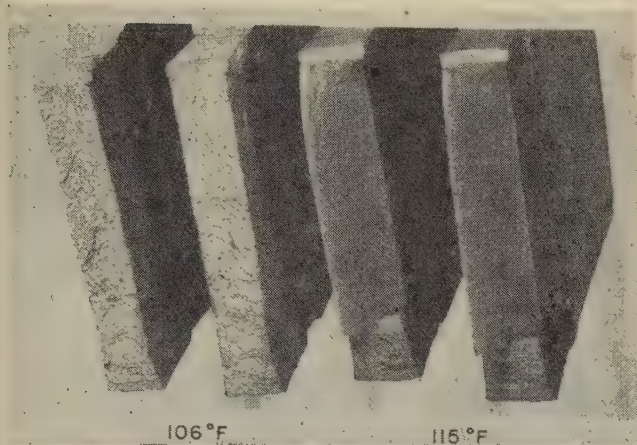
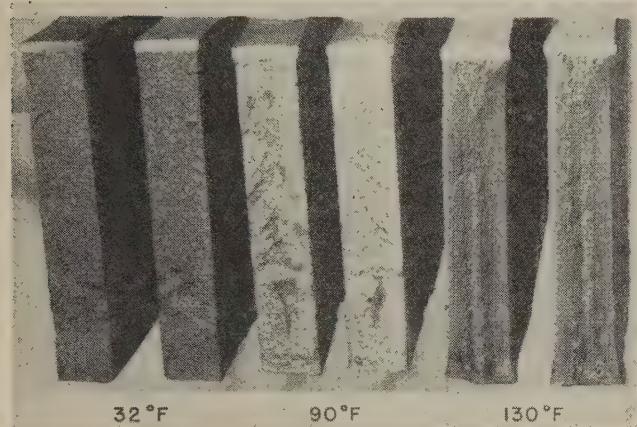


Fig. 15 (a)



Figs. 15 (b)

FIG. 15 (a) AND (b) FRACTURES IN RIMMED STEEL

(Fractures, shown in *a*, in rimmed Steel 7 obtained by cleavage-tear tests just below and above transition temperature. Fractures, shown in *b*, in rimmed Steel 7, obtained by cleavage-tear tests at temperatures indicated. Below transition-temperature range, fractures are of cleavage type, and those above this range are of shear type.)

unmistakable. Numerous tests made on various types of steel not related to this investigation have confirmed the latter.

Fig. 15 is a graphical presentation of the transition temperatures of the steels in the as-received condition, as determined by the conventional Charpy impact and cleavage-tear tests. The difference between the transition temperatures of the four steels determined by these two test methods is clearly evident. It is also evident that a sharp change occurs in the unit breaking loads at the transition temperature of each steel, higher loads being required by the shear or ductile fractures and lower loads by the brittle or cleavage fractures.

It is to be noted from Fig. 15 that the steels are classified in the same order by the cleavage-tear and impact tests, with respect to their transition temperatures. This appears to prove that the "notch sensitivity" of a steel can be evaluated effectively without measuring the specific work done or the energy absorbed by fracturing, as is the practice in impact or other tests, but by merely measuring the breaking loads, in pounds per square inch, obtained with the use of a suitable test coupon. The latter method is simpler, it yields more accurate results since nonhomogeneity and similar factors are eliminated by the use of large test coupons, and its results instead of being in foot-pounds are in terms of P/A , which the designer can use.

On account of the excessive deflections obtained in shear tests,

no attempt is made to apply Equation [4] for computing breaking loads for the latter type of fractures, since the correction factor v in the equation is a variable. However, the data listed in Table 11 show that about 30–45 per cent higher loads are required to produce shear fractures than cleavage fractures. The P/A values for cleavage fractures occurring below the transition-temperature range conform to the values computed by Equation [5], as has been shown in previous sections of the paper.

It is beyond the scope of this paper to discuss the validity of the assumption that the transition temperature of a given steel is a fixed temperature range, regardless of the states or systems of stress to which the section of that steel is subjected. It is apparent that the conventional impact tests are of qualitative, rather than quantitative value. Service failures of steel plates in monolithic structures seem to indicate that their transition temperatures under service conditions are at least about 100 F higher than those shown by impact tests, and tests recently made on fairly large models of structures (6) have confirmed this. Since the transition temperatures determined by the cleavage-tear tests are also about 100 F higher than those indicated by the impact tests, it appears that the results of the cleavage-tear tests described in this paper are more nearly indicative of the behavior of a given steel under service conditions than those obtained by the conventional impact tests. Transition temperatures under service conditions are further discussed in the paragraph immediately following.

As the test results reported in previous sections of this paper show, it is necessary to use a test coupon whose depth is at least 6 in. in order to obtain an approximate state of balanced stresses, and thereby to minimize excessive bending and deflection. It has also been shown by the data in the notch-radius section of the paper that it is necessary to use notches whose root radii are smaller than those of the notches of the standard Izod and Charpy impact-test coupons, in order to obtain stress concentration of an order sufficient for initiating cleavage fractures from the notch. The transition-temperature ranges determined by the cleavage-tear tests reported in this section were obtained by the use of coupons which conform to these requirements and have a 45-deg notch with a root radius of 0.0015 in.

Exploratory tests have shown that if a notch is used whose root radius is smaller than the latter, the transition-temperature range established would be somewhat higher than those reported, and that if the notch root radius is made as large as the Charpy notch root radius (0.039 in.), the transition-temperature range found would be somewhat indefinite and considerably lower. More data are needed to determine whether or not a leveling off occurs in this relationship at some value of notch root radius or of temperature. The notch-root-radius dimension, as has been shown, is one of the main factors which determines the state of stress. It would appear then that the state of stress determines the level of the transition-temperature range and that, unless the state of stress is defined, the transition-temperature range cannot be defined for a given metal.⁴

DISCUSSION

In monolithic structures of all-welded construction, multiaxial stresses develop under service conditions, and in such structures eccentric loading cannot be avoided in most cases. Some small notches such as weld underbead cracks in some segregated zones, slight weld undercuts, or shallow fissuring or discontinuities in the parent plate may be present, which may escape detection in the finished structure, even with careful inspection. Massive

⁴ The determination of transition temperatures and correlation of the test results with service conditions are treated in another paper by the author, published in the *Welding Journal*, vol. 27, 1948, Research Supplement, pp. 123-s–131-s.

castings and forgings fall in the same category. These irregularities constitute notches from which cleavage fractures may develop under multiaxial stress conditions and therefore may be considered conditions encountered in service which are very similar to those designed in the cleavage-tear test used in this investigation.

The test data presented in this paper show that mild steels may fail by development of a cleavage or brittle fracture, by the ductile or shear fracture, or by a combination of both fractures, depending upon the conditions. The controlling conditions are the extent of restraint to the flow of the metal in directions transverse to the fracture, and the temperature of the metal subjected to fracturing. The speed of load application is another factor if

appreciable deformation occurs or is possible under the state of stress developed. The welds tested were found to behave similarly to steel plates. The cleavage fracture is characterized by its "sugary" or "crystalline" appearance, by the absence at the fracture of an appreciable ductility such as stretching or deformation, and by its sudden appearance at a given load level and rapid rate of penetration. It has been stated that cleavage fractures have a rate of propagation of about 3500 to 4500 fps (7).

The shear fracture on the other hand has a "silky" appearance, shows evidence of normal deformation, such as elongation and area reduction, its rate of propagation is slow, and the extension of its fracture gradual. For the sake of simplification, it has been stated that cleavage fractures develop by separation of the

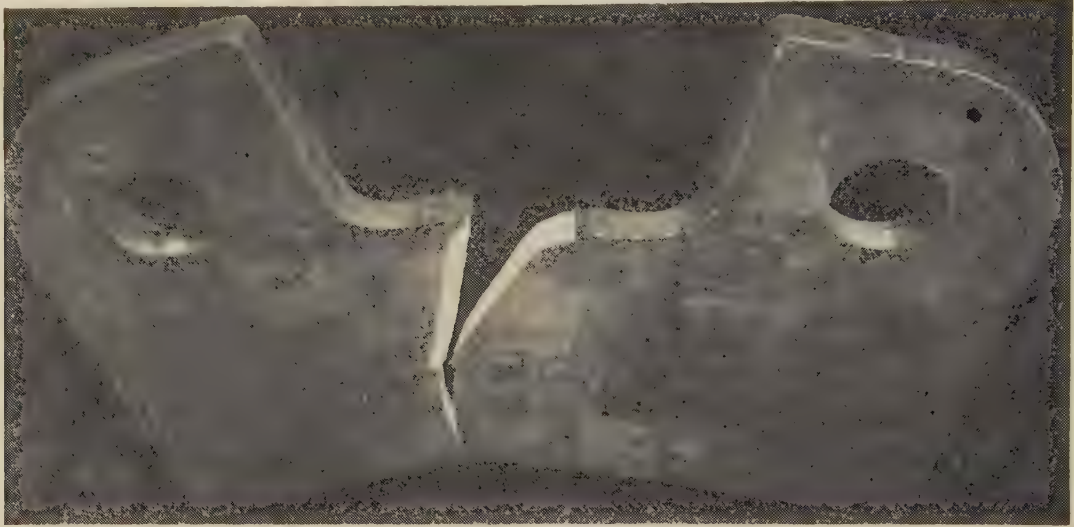


FIG. 16 APPEARANCE AND COURSE OF DOUBLE SHEAR FRACTURE WHICH STARTED FROM A 120-DEG NOTCH

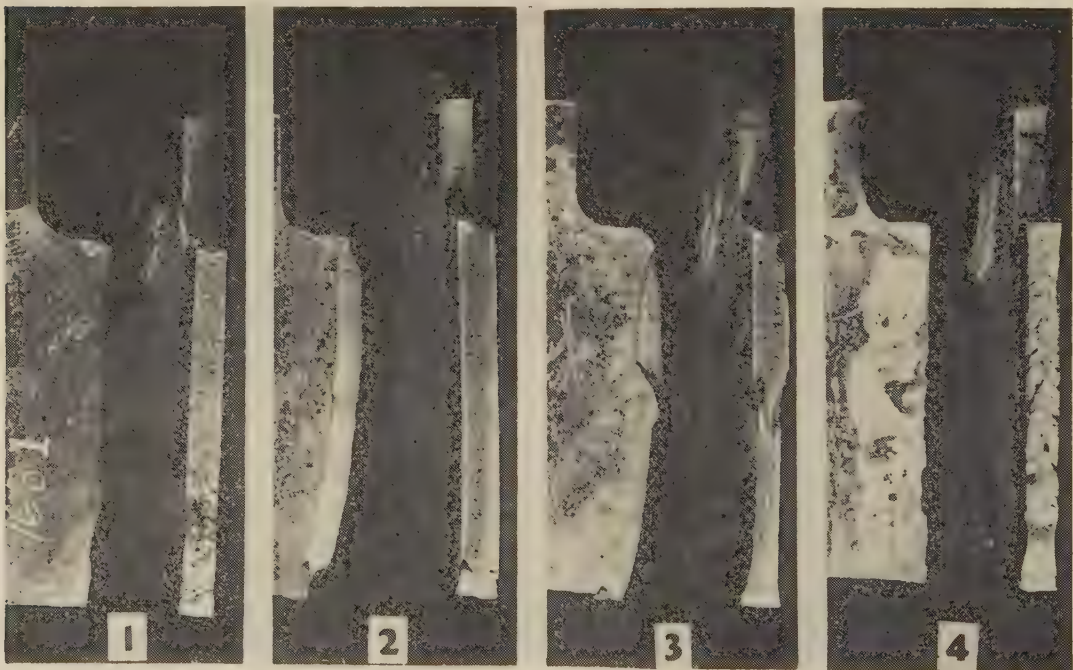


FIG. 17 TYPES OF FRACTURES

(Fracture in view 1 is of cleavage type, those in 2 and 3 are shear type. In view 4 main fracture is of cleavage type, with a narrow shear fracture at edges, the latter running at an angle of 45 deg with plane of main fracture.)

metal along certain cleavage planes, whereas shear fractures develop by sliding of one part over the other to the point where separation occurs.

Several types of shear fractures produced by the cleavage-tear tests were encountered in this investigation, the orientation and nature of some of which are shown in Figs. 16 and 17. Fig. 16 shows a double-shear fracture which runs at an angle of about 45 deg with the notched edge of the test coupon. The fracture in Fig. 17, view 1, is cleavage type, and was included for comparison. The coupon in Fig. 17, view 2, shows a shear fracture, which runs at an angle of about 45 deg with the face or vertical plane of the coupon. In Fig. 17, view 3, the fracture starts at the same angularity as in view 2, but rotates itself through an angle of 180 deg, as the center of the test coupon is reached. Figure 17, view 4, shows slight shearing along the edges of the cleavage fracture, the course of the shear fracture being at an angle of about 45 deg with the plane of the main cleavage fracture. These shear fractures have been described here somewhat in detail, because they confirm that the stresses involved in the test coupons used are multiaxial in nature. This has also been confirmed by stress measurements made simultaneously with these tests, the results of which, however, not being sufficiently classified, are not presented in this paper.

The fracture shown in Fig. 18(a) is 16 in. in length and was produced in a coupon of that depth. Upon application of load, a cleavage fracture was formed, penetrating to a depth of about 2½ in. below the notch. It was arrested at that point by an interposing shear fracture, the latter starting from both faces of the coupon, in a manner similar to the shear fractures seen along the edges of the coupon in Fig. 17, view 4, and gradually extended itself to the center of the coupon. The latter finally resulted in a narrow band of V-shaped shear fracture, which acted as a barrier against the progress of the cleavage fracture. The formation of this shear fracture probably occurs in the same manner as that often found along the edges of conventional tensile-test fractures. When the load was then reapplied, the entire remaining section broke by a single cleavage fracture. The contrast between the shear and brittle fractures is seen in Fig. 18(a), the V-shaped shear fracture being somewhat darker due to its finer appearance. Fig. 18(b) shows this section in natural size, in which it is clearly seen that considerable contraction has occurred at the area where the shear fracture predominates.

Typical cleavage fractures in 22-in.-deep coupons are shown in Fig. 19. One of the characteristics of cleavage fractures is clearly shown in actual size in Fig. 19(b), this characteristic being a herringbone appearance given to the fracture by small ridges and valleys formed by the sudden rupture. A comparison of this herringbone pattern with that of the shear pattern shown in Fig. 18 reveals that the former points toward the origin of the cracks whereas the latter points in the opposite direction. Cleavage fractures which have been arrested by shear fractures, similar to that shown in Fig. 18, have occurred in a few all-welded ships, the termini of which cracks, upon being chipped for repair, have been found to penetrate 1 to 2 in. deeper in the center than at the surface of the plate. The herringbone pattern of the cleavage fractures just referred to has been found to be of considerable aid in tracing fractures to their origins.

The characteristic difference between the shear and cleavage fractures is more clearly seen under the microscope. Fig. 20(a) shows at a magnification of 100 diam the severe distortion and cold-working present in the metal adjacent to a shear crack, and Fig. 20(b) shows a portion of the same fracture at a magnification of 1000 diam, with more details of cold-working and deformation being discernible at the latter magnification. Fig. 20(c) shows, likewise at $\times 1000$, a cleavage fracture merging into a shear fracture. In this case considerable shattering and extension of cracks

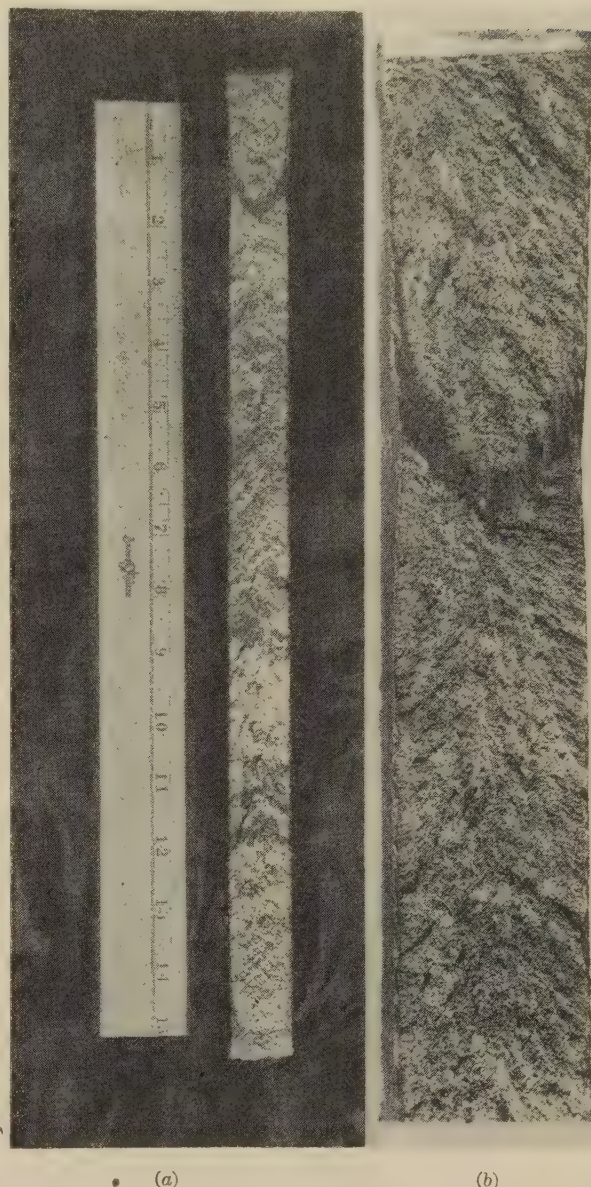


FIG. 18 CONTRAST BETWEEN SHEAR AND CLEAVAGE TYPE OF FRACTURES

(V-shaped and narrow band of fracture of dark appearance slightly below upper edge of coupon is a shear fracture, the rest being cleavage fracture. Deformation associated with shear fractures is depicted by b, in natural size. Note slight but definite contraction in section of plate, approximately 2½ in. below notch, where shear fracture occurred.)

in many directions are observable. A typical cleavage or brittle fracture is shown in Fig. 20(d), also at $\times 1000$. It is to be noted that the crack in the latter case is straight, penetrates ferrite and pearlite grains indiscriminately, and terminates with a very sharp end. Fractures of the latter type indicate that they are produced by a triaxial state of stress of high order.

Numerous measurements have shown that the radius at the terminus of cleavage fractures is not the same in different fractures, the smaller radii often being found in fractures formed at low temperatures and by loading systems which release a high potential energy when fracture occurs. The smallest radius found was of the order of three-millionths of an inch. This has been referred to in Table 8, in connection with the data on the effect of notch radius on breaking loads. It is apparent that the radius at the base of these cleavage fractures, although very small,

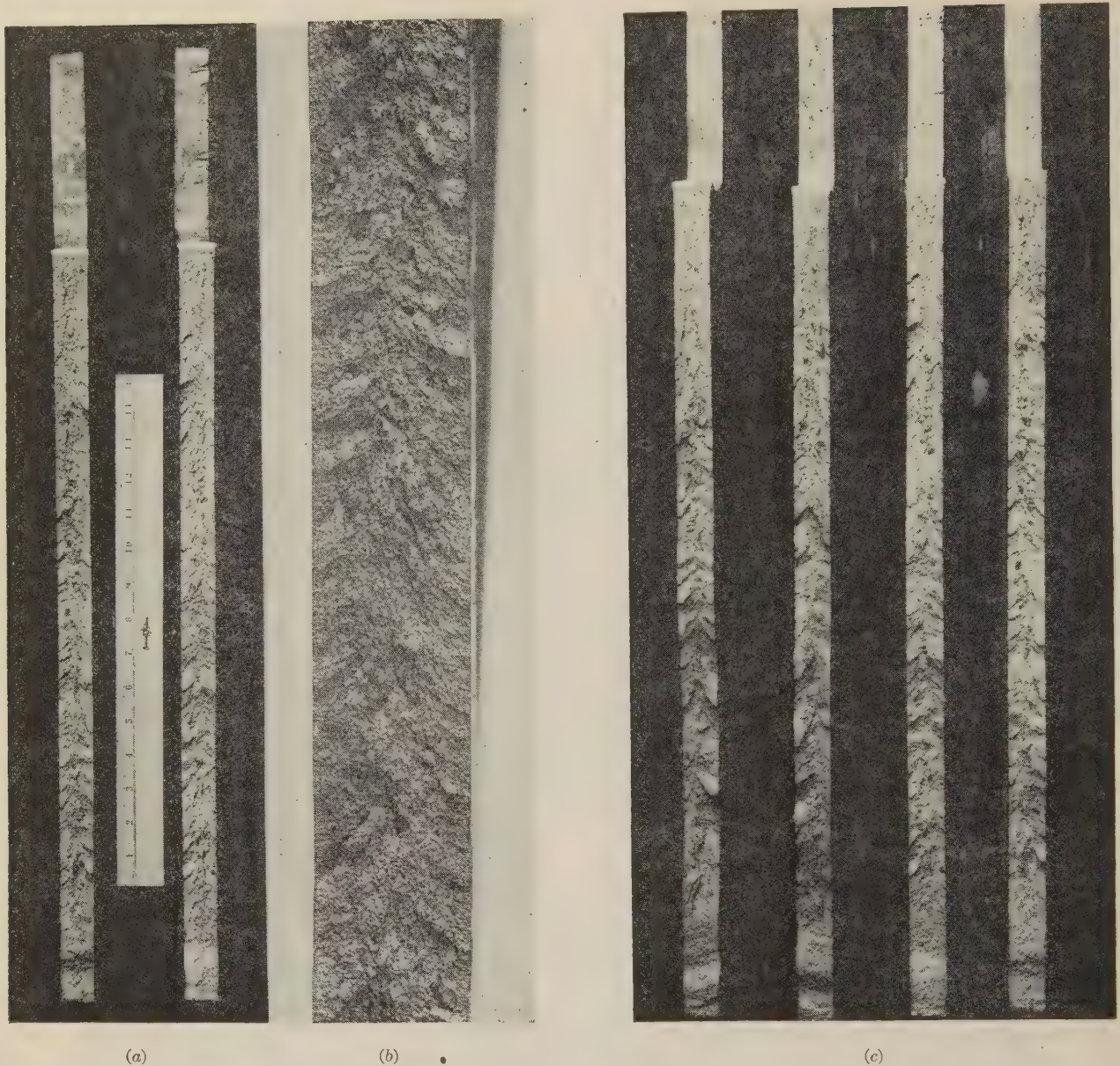


FIG. 19 TYPICAL CLEAVAGE FRACTURES

(Typical cleavage fracture, shown in *a*, of 22 in. depth obtained in steel 17. Fracture developed suddenly at testing temperature of 32 F with a sharp retort, at a load of 268,400 lb, the entire section breaking into two pieces. The coupon was 22 in. deep, 1 in. thick, with an eccentricity of loading of $12\frac{1}{2}$ in. It had a 45-deg notch of $\frac{3}{16}$ in. depth, with 0.0015 in. radius at base of notch. Unit breaking load amounts to about 12,200 psi. *b*, Shows in natural size, portion of same fracture. Note herringbone formation in this fracture, which is typical of cleavage fractures. Arrow heads point toward origin of fracture. *c*, Cleavage fractures developed in 22-in-deep coupons tested at 32 F, the notch used being 45 deg and $\frac{3}{16}$ in. deep.)

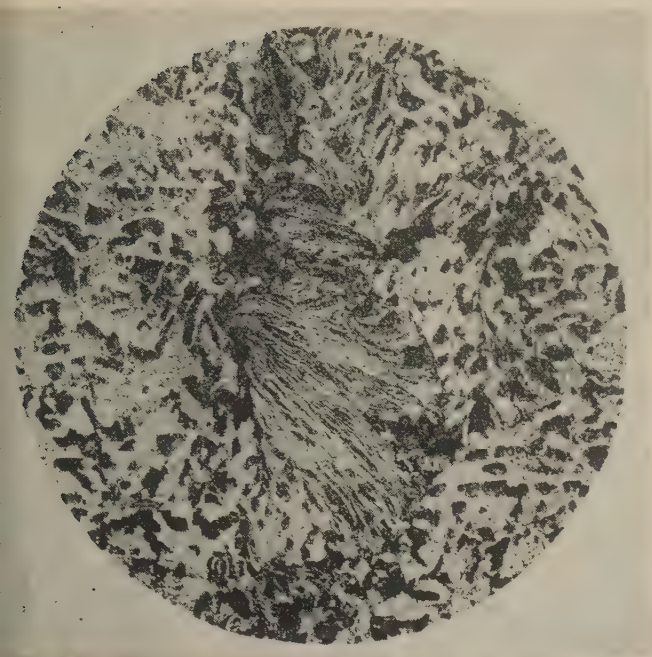
has a finite value. Otherwise, as has been discussed previously and as Equation [14] shows, the load required to extend such fractures would equal zero which, obviously, is not true.

The shear or ductile fracture is less dangerous because it is always associated with deformation which requires expenditure of a large amount of energy. Unless energy is available at the required rate to continue the fracture, the extent of damage of this type of fracture is localized. The cleavage fracture on the other hand occurs with practically no deformation and hence with no appreciable amount of energy absorption. The latter type of fracture appears suddenly as soon as a certain load level is reached, progresses with a high rate of penetration and causes more extensive damage, because only a slight amount of energy is dissipated by this type of fracturing.

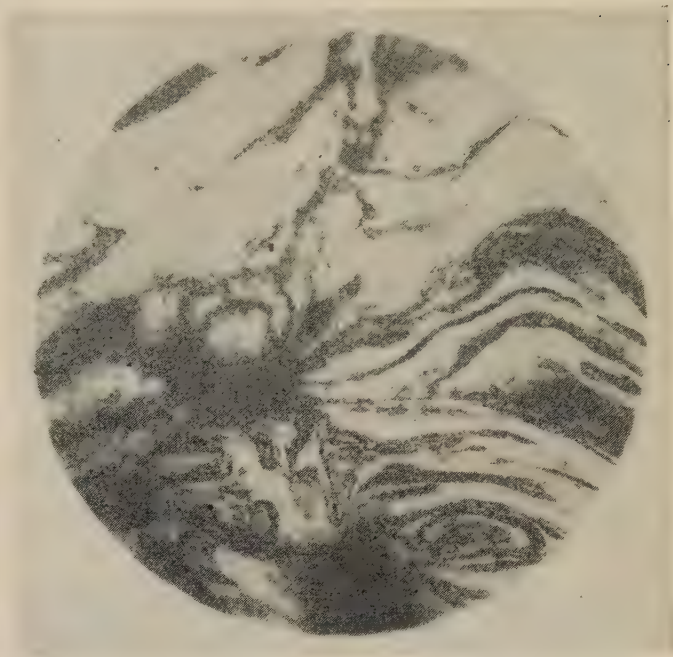
As has been shown, the cleavage or brittle fractures are the normal rather than the abnormal type of fractures that occur

under a certain set of conditions, whereas the ductile or shear fractures are the normal type producible under another set of conditions. There are overlapping conditions in which a combination of both fractures may occur. As stated, development of brittle fractures is favored by increasing the restraint to transverse flow, by decreasing the temperature, and by increasing the rate of load application, the latter being a factor only if appreciable deformation is obtainable. Unfortunately, however, these factors can seldom be controlled under service conditions. Therefore, the damaging effect of cleavage fractures can be minimized by other means, such as by modification in design, or by selection of better materials of construction, or by a combination of both.

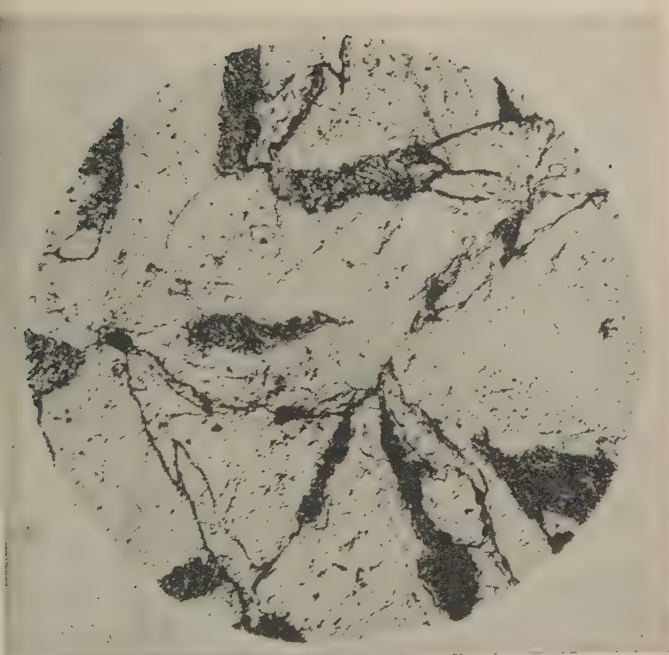
Of the various design modifications suggested, the one that has found most favor and has been put into practical use to a certain extent consists in inserting riveted seams in some of the critical sections of monolithic structures, such as all-welded ships. The



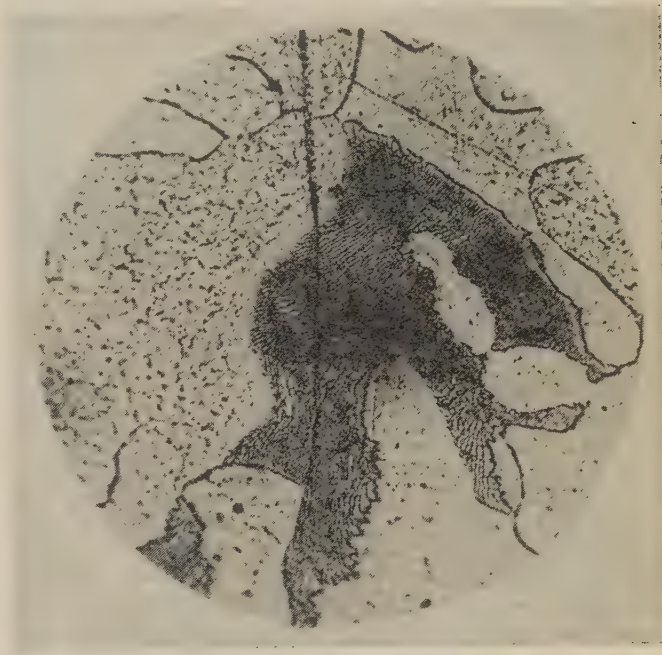
(a)



(b)



(c)



(d)

FIG. 20 MICROSTRUCTURES OF TYPICAL FRACTURES

(a. Shows severe deformation in shear fracture. Etchant, picral-nital; $\times 100$. b. Shows portion of same shear fracture, as seen at higher magnifications. Etchant, picral-nital; $\times 1000$. c. Shows merging of a cleavage fracture into a shear fracture. Note shattering and extension of cracks in many directions. Etchant, picral-nital; $\times 1000$. d. Terminus of typical cleavage fracture. Note that distortion is absent and that crack penetrates ferrite and pearlite grains indiscriminately. Crack apparently stops abruptly, with a sharp radius and without any offshoots. Etchant, picral-nital; $\times 1000$.)

object of this modification is to break up the monolithic structure into several independent sections in order to obtain structural flexibility and also to provide discontinuity in the plates so that when a cleavage crack starts, it will terminate at the end of one of the plates without continuing itself into the next or overlapping plate.

The test data presented in this paper show that the external load required to break a given section, under eccentric loading and multiaxial-stress conditions, increases as the radius at the base of the crack or angle of the notch increases. On this basis, in order to arrest the progress of cleavage cracks, it is necessary only to increase the notch angle and notch radius produced by the crack. This can be done simply by drilling holes of certain size in the sections in question, so that when cleavage fractures develop they will terminate into one of these holes. The continuity of the crack is abruptly terminated at such holes, since there both the notch radius and notch angle are increased. The latter are determined by the curvature of the hole used. It is obvious that in order to accomplish this no rivets are required, except that where necessary some arrangement should be made to prevent leakage. To obtain the latter, butt straps could be applied over the perforated area, either externally or internally to the vessel, or both, thereby, in addition to preventing leakage, a reinforcement is provided for the section weakened by the holes. If desired, the holes could be drilled in the butt straps only, instead of in the shell plates, and these butt straps installed by welding at the desired sections and in the desired direction. It is believed that the holes in the butt straps will have the same effect for arresting the progress of the cleavage or brittle fractures as the ones drilled in the shell plate.

Experimental data show that the actual diameter of the holes drilled need not be larger than one half the shell-plate thickness. In most cases these holes could be closed by filling them with a suitable material to safeguard against leakage if it is not necessary to reinforce the section by butt straps. Tests have demonstrated that holes effectively stop extension of cleavage fractures. Figs. 21(a) to (d) show several variations of applying the foregoing principle, and illustrate how effective these holes are in arresting extension of cleavage fractures.

The use of a material of construction for ships and pressure vessels which would resist propagation of cleavage fractures, undoubtedly has some outstanding advantages. As stated previously, the conventional tensile tests and the impact tests do not give quantitative indications regarding the behavior of a given steel in terms of its resistance to cleavage-fracture development. On the other hand, with the cleavage-tear test a more quantitative determination of the foregoing is possible. The deoxidized steel 16, for example, was found to be distinctly superior to the other steels in resisting extension of cleavage fractures, especially in certain heat-treated conditions. Obviously, the use of a "sub-zero" temperature steel for construction of the critical sections of a ship or pressure vessel would reduce materially the hazard of cleavage-fracture propagation. If such a steel is used, any crack originating from some discontinuity present in the structure is more likely to be the shear type rather than the cleavage or brittle type. Specially deoxidized and denitrided steels (8) of fine grain structure would undoubtedly be economical for certain critical applications. In some cases, such steels could be used in conjunction with the design modification described in the preceding paragraph.

One of the peculiarities of cleavage fractures consists in the fact that their depth of penetration cannot be predicted without knowing the state of stress and possibly many other variables, including metallurgical variables of the section subjected to fracturing. The P/A value may be the same for two test coupons, in one of which the crack depth may extend the full section, but

only a short distance in the second one. This is illustrated by Fig. 18(a) in which the load required to produce a crack 2 in. in length in a 16-in-deep test coupon was found to be the same as that required to produce a crack which penetrated through the entire section of another test coupon of identical dimensions, geometry, and material specifications.

In the former case, the crack was arrested by a shear fracture, whereas in the latter, cleavage fracture persisted through the entire depth of the coupon. Aside from the metallurgical variables, the depth of crack penetration then is determined by the stress pattern, several other variables, and by the level of the potential energy that has to be reached under a given set of conditions before rupture results. Herein lies the main difficulty of evaluating materials on the basis of fracture work, since the variables involved are often difficult to evaluate or identify. Determination of unit breaking loads, on the other hand, constitutes a better index for evaluating resistance to fracturing as has been shown. The P/A values would in addition indicate whether the fracturing is to occur by shear or by cleavage. The transition-temperature tests reported in an earlier section of the paper support this.

The test data appear to show that the load at which yielding occurs can be raised almost to coincide with the breaking load of the steel by the use of a proper coupon, notch geometry, and testing conditions. This is indicated by the close agreement of the experimental data with theoretical equations made use of on the assumption given, even though a few modifications were made in these equations in order to satisfy the peculiarities revealed by the tests.

On the basis mentioned, it would appear possible to evaluate, approximately, the load-carrying capacity of steel under certain conditions, by the use of the derived equations. As an example, let us attempt to compute for centric and tangential loading conditions the nominal breaking or cleavage strength transverse to the rolling direction of a steel beam of rectangular cross section with a net depth of 6 in., or greater and, say, $3/4$ in. in thickness, which section contains in one of its edges a crack $3/16$ in. in depth and at right angles to the direction of load application.

In this case the general Equation [15] can be used. It has been shown that from a practical standpoint 6 in. could be substituted for the term $(d - h)$ in the equation to represent section depths greater than 6 in. It has also been shown that for cracks the angle of notch can be assumed to be zero, and the notch radius at its base equal to about 3×10^{-6} in. By substituting these values of d , h , N , and R in Equation [15], the following simplified equation is obtained

$$\frac{P}{A} = \frac{23,700}{1 + 0.617e} \quad [16]$$

Equation [16] applies to fractures transverse to the rolling direction of the plate. The equation applying to fractures parallel to the rolling direction, derived by modifying Equation [16] by Equation [8] is as follows

$$\frac{P}{A} = \frac{23,700}{1 + 0.752e} \quad [17]$$

For centric loading $e = 0$, and therefore the value of P/A is 23,700 psi, according to Equation [16], which value is the same as that computed by Equation [15] and discussed under the section dealing with notch radius.⁵ For tangential loading, $e = (6 + 3/16)/2$ or approximately 3.1 in., substitution of which in Equation [16] yields the value of $P/A = 8150$ psi for the latter type of

⁵ For sections greater than 6 in. in depth the value of e found by the relation $e = (d - h - 6)/2$ is equivalent to the value of $e = 0$ for the 6-in-deep coupon (Equation [10]).



FIG. 21 (a) CLEAVAGE CRACK WHICH PENETRATED ENTIRE DEPTH OF STEEL COUPON $\frac{3}{4}$ IN. THICK $\times$ 24 IN. DEEP (It formed a secondary crack about $8\frac{1}{2}$ in. below notch. Entire section broke with a snap. Testing temperature: 0 F.)



FIG. 21 (b) ARREST OF CLEAVAGE CRACK BY A $\frac{3}{4}$ -IN. HOLE AT 8 IN. FROM BOTTOM (With exception of holes, geometry of coupon and notch and testing conditions were the same as in Fig. 21a.)

loading. The foregoing are for fractures transverse to the rolling direction. For fractures parallel to rolling, the corresponding values for centric and tangential loading are 23,700 and 7100 psi, respectively, as computed by Equation [17].

If we assume that in the cases cited the service temperatures are above the transition temperature range of the steel, and that as a result fractures would be of the shear type, the breaking loads would be 30-45 per cent higher than the values given. It

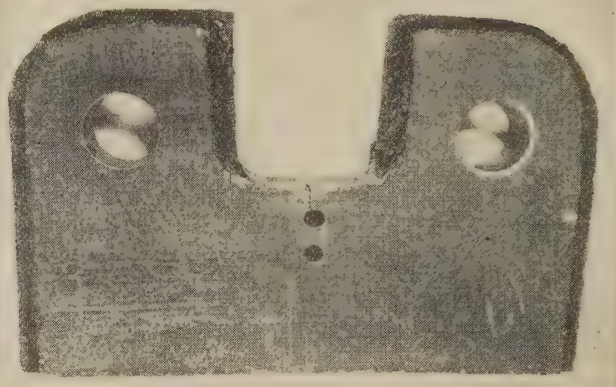


FIG. 21 (c) THE 1-IN-THICK, 6-IN-DEEP COUPON IS "ENVELOPED" IN TWO $\frac{1}{2}$ -IN-THICK BUTT STRAPS APPLIED ON BOTH FACES OF COUPON BY WELDING AT EDGES (Two holes of $\frac{3}{8}$ -in. diam were drilled in each butt strap below notch. Crack penetrated 1-in. plate to some distance but was finally arrested by holes in butt straps.)

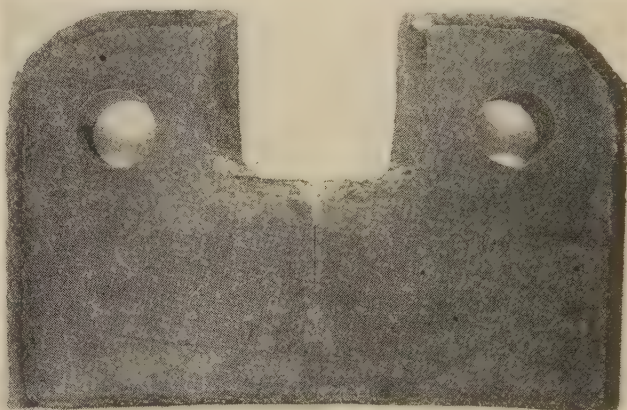


FIG. 21 (d) SAME AS FIG. 21 (c) EXCEPT 1-IN-THICK CENTER PLATE WAS DRILLED INSTEAD OF BUTT STRAPS (The crack was arrested at hole in center plate.)

has been shown previously that the maximum notch effect is obtained by notch depths of $\frac{5}{32}$ to $\frac{3}{16}$ in., and that the effect of mass is not appreciable beyond a section depth of about 6 in. Therefore the values of P/A are not changed materially by increasing the depth of the section or of notch if proper correction is made to compensate for the change in loading eccentricity resulting from increasing the notch depth, in accordance with Equation [15].

In the case just given the thickness of $\frac{3}{4}$ in. was used in order to assure that the section is sufficiently thick to support development of cleavage fractures, although we have produced cleavage cracks in D-type coupons made from $\frac{1}{4}$ -in-thick steel plates. The unit breaking load would not be expected to change if the section was assumed to be thicker than $\frac{3}{4}$ in.

The equations are applicable to mild steels, the tensile strength of which is approximately 60,000 psi, since steels of the latter tensile strength were used in deriving the data. Exploratory tests show, however, that they could also be applied to steels of somewhat higher or lower tensile strengths, if the equation used is modified by the ratio of the tensile strengths. It is to be noted that Equation [15] covers only the geometrical variables of the notch and of the coupon, the effect of which variables on the breaking loads of the steels tested was found to be essentially the same. Although the terms covering the effects of temperature and of response to heat-treatments are not included in Equation [15],

they are identified and can be evaluated by the cleavage-tear tests described.

A confirmation of the unit breaking-load figures of 7100 to 8150 psi for tangential loading as discussed is obtained by comparing them with the results of an entirely different type of test. The breaking loads of mild steels under corrosion-fatigue conditions or with notched bars are found to be not too far off from the given figures if the details of the test conditions are considered (9). In most of the latter type of tests the stresses involved are of the tangential type, and the notches used, or the pitting and intergranular deterioration produced by corrosion, create conditions which are almost identical with those of the case discussed.

It appears from these figures that if cracks as shallow as $\frac{5}{32}$ to $\frac{3}{16}$ in. are present in certain orientation in critical sections of a monolithic structure, our current values of factors of safety of 3 to 5 need reconsideration in order to safeguard against continuation of such cracks to the point of ultimate failure of the structure, even if the loading is within the section in question. This is obvious, since the nominal tensile strength of 60,000 psi, determined by the conventional tensile tests and normally used as basis for design, is drastically reduced as shown, by stress-concentration and other factors induced by the presence of notches, cracks, or similar discontinuities.

If the allowance of factors of safety of relatively high numerical values becomes impractical by design or other peculiarities, then the design or material modifications, or both, could be considered. The latter may be found to be more economical in most cases. Monolithic structures of all-welded construction, such as welded ships, certain types of pressure vessels, and similar structures are particularly susceptible to developing large cleavage fractures from some obscure discontinuity, and deserve the precautions mentioned as to design and selection of materials.

Inasmuch as the load-carrying capacity is a criterion of prime importance, the correlation of the P/A values with several of the variables including section and notch geometries, temperature, etc., should have some practical usefulness. The experimental data show that two distinct types of fracture and fracture strength are to be recognized, one being the brittle or cleavage type manifesting slight deformation and lower nominal breaking strength, and the other ductile or shear type, exhibiting ductile fractures and considerably higher breaking loads. It is believed that the test data presented in this paper show that both of these types of failure appear to follow certain patterns and laws, and that they can be evaluated properly and safeguarded against. However, an immense amount of work needs to be done in this field in order to solve the mechanism of fracturing of metals in all its intricacies.

SUMMARY AND CONCLUSIONS

1 A test coupon and procedure have been developed and described for determining the tensile breaking loads of edge-notched rectangular steel sections, and the conditions under which failure occurs by cleavage fracturing. Several grades of ship-plate and pressure-vessel steels of rimmed, semikilled and killed qualities, covering a thickness range of $\frac{1}{2}$ to $2\frac{1}{4}$ in., and several sections of butt welds made by manual and submerged-arc processes were investigated. Cleavage fractures were developed in all of these.

2 The effects of eccentricity of loading, section depth and thickness, notch geometry, temperature of testing, rate of loading, and heat-treatments were studied and their effects on the breaking loads determined.

3 Two distinct modes of failure of steel were encountered, one being by the cleavage or brittle fracturing, and the other by the ductile or shear fracturing. The nominal breaking strength of steel by cleavage fracturing was found to be approximately the same for steels of the same tensile strength, and appreciably lower than that by shear fracturing.

4 The presence of a notch of proper geometry and in proper orientation appears to expand the elastic range of steel and to raise the proportional limit almost to coincide with the breaking point of the steel. By the use of modified equations the breaking loads can be computed for notched rectangular sections, if the sections are considered as a whole, without evaluating the stress at the notch. These results are obtained if the temperature of the steel is below its transition temperature range.

5 The state of stress established by a notch of the type cited establishes a well-defined tensile breaking load for the section containing the notch. If the load is eccentrically applied, the unit breaking load for sections greater than 6 in. depth is approximately the same as that for the 6-in. section, with equivalent loading eccentricities. It appears that a critical depth of section is required to compensate for the stress gradients established by the notch, and also to resist bending and deformation of the section.

6 Steels exhibit greater resistance to propagation of cleavage fractures in the direction transverse to rolling than in the direction parallel to rolling of the plate. The anisotropy is clearly revealed by the cleavage-tear tests described in this paper, if loading eccentricities greater than tangential are used.

7 The rate of loading within the rates of 0.06 to 2.2 ipm had no effect on unit breaking loads required to produce cleavage fractures.

8 The effect of the notch-geometry variations, including depth, angle, and root radius of the notch, was found to be of a parabolic type, indicating certain limits beyond which effects of these variables are not appreciable.

9 The notch depth or notch severity does not appear to be an important factor in so far as large sections are concerned, the optimum effect being produced by a $\frac{5}{32}$ to $\frac{3}{16}$ -in.-deep notch. Even a 45-deg notch of $\frac{1}{32}$ in. depth is sufficient to produce cleavage cracks in a steel section 1 in. thick and 6 in. deep, if the radius at the base of the notch is small, say, of the order of 0.0015 in. or less.

10 A finite load is required to continue the extension of a crack into the sound structure, this load decreasing rapidly as the notch radius is decreased. Incipient fissures and similar discontinuities in the surface of steel plates, forgings, and of castings, and weld underbead cracks, etc., are therefore dangerous. It is estimated that the relative stress concentration at the base of cleavage cracks is in excess of 9, if the average stress of the section is assumed to be unity, and the actual stresses required to initiate fracturing in the presence of a notch probably approach the technical cohesive-strength or the actual cleavage strength values of the material.

11 Normalizing and especially water-quenching increase the resistance of steels of the type tested to development of cleavage fractures. Annealing of steel produces erratic results and appears to be somewhat harmful in so far as resistance to cleavage fracturing is concerned. Stress-relieving of welds at 1200 F can, in general, be considered helpful for increasing the resistance to cleavage fracturing.

12 The transition temperatures of steels were determined by the cleavage-tear tests, by measuring unit breaking loads expressed in pounds per square inch. The transition temperatures found by these tests for several 1-in.-thick steel plates are 85–100 deg F higher than those indicated by the Charpy impact tests, and therefore appear to be more nearly indicative of the behavior of steel in service. Nevertheless, the transition temperature of a steel cannot be defined under service conditions unless the state of stress that prevails under the same conditions is also defined.

13 A given section of steel may fail by shearing or by cleavage fracturing, depending upon the state of stress involved, temperature of the metal, and the condition of the material, each type of

fracture being normal within certain limiting conditions. The behavior of the welds tested was found to be very similar to rolled steel plates, with respect to development of cleavage and shear fractures. In general, notches 90 deg or smaller and notch-root radii smaller than about 0.01 in. favor development of cleavage fractures.

14 Shear fractures are less damaging because of the resultant deformation, which localizes the extent of their damage, and their rate of propagation is slow. Cleavage fractures, on the other hand, produce practically no deformation, occur as soon as a critical load level is reached, and propagate at a very rapid rate.

15 If a small crack or a discontinuity similar to a notch is present or is developed in a monolithic structure in service, large cleavage fractures are likely to develop from it, unless the imposed loads are low. Approximate computations made on the basis of the data presented in this paper indicate that our present factors of safety are inadequate to safeguard against possible failures originating from notches or discontinuities.

16 Several remedial measures are suggested for minimizing the potential damage of cleavage fractures in monolithic structures and pressure vessels. These suggestions cover modifications in design and in material of construction.

ACKNOWLEDGMENT

The author is grateful to J. N. Pew, Jr., of the Sun Oil Company management, for his guidance and encouragement in this work. Special thanks are due to several of the metallurgical employees of the Sun Oil Company, who worked on this project, especially to J. E. Carney for valuable assistance and for developing the notching technique used, and to H. N. Seward for conducting most of the tests and preparing the photographs. The co-operation given by the Sun Shipbuilding and Dry Dock Company is gratefully acknowledged. Thanks are also due to H. W. Graham, of Jones and Laughlin Steel Corporation, and his staff, for comments and for conducting considerable exploratory tests on several grades of steel by the cleavage-tear testing methods; also to Lukens Steel Company for furnishing several sections of a titanium-treated steel plate for testing.

BIBLIOGRAPHY

- 1 Numerous reports covering investigations and researches sponsored by the Bureau of Ships, Navy Department, and by other governmental agencies.
- 2 "Slow Bending Tests on Large Notched Bars," by J. G. Dacherty, *Engineering*, London, England, vol. 139, Feb. 22, 1935, pp. 211-213.
- 3 "The Influence of Size on the Brittle Strength of Steel," by H. N. Davidenkov, E. Shevandin, and F. Wittmann, *Journal of Applied Mechanics*, Trans. ASME, vol. 69, 1947, p. A-63.
- 4 "Plasticity," by A. Nadai, McGraw-Hill Book Company, Inc., New York, N. Y., 1931.
- 5 "Report on Behavior of Ferritic Steels at Low Temperatures," parts I and II, by H. W. Gillett and F. T. McGuire, ASTM, 1945.
- 6 "Some Tests of Large Welded Structures," by E. P. DeGarmo, J. L. Meriam, and R. C. Grassi, *The Welding Journal*, vol. 26, 1947, pp. 267-267.
- 7 "Some Causes of Brittle Failures—Welded Mild Steel Structure," by H. E. Kennedy, *The Welding Journal*, vol. 24, 1945, pp. 588-598.
- 8 "Effect of Deoxidation on the Strain-Sensitivity of Low-Carbon Steels," by H. K. Work and G. H. Enzian, AIME Technical Publication No. 1835, 1945.
- 9 "Prevention of Failure of Metals Under Repeated Stress," by Battelle Memorial Institute Staff, John Wiley & Sons, Inc., New York, N. Y., 1941.

Discussion

J. G. ALTHOUSE.⁶ The work done by the author is of great

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interest and will contribute to the available knowledge regarding the physical characteristics of steel under certain conditions of stress. An effort was made to obtain relative results on different types of steel by the use of a new type of test that might be described as a combination notched tension bend test.

The use of this test appears to eliminate some of the variables which influence other proposed tests for the determination of a transition temperature where the steel develops a cleavage or brittle fracture from a so-called shear or ductile fracture. Considerable difficulty is encountered in trying to compare the test results on small specimens with the actual results from large welded structures. The evaluation of a steel quality, as to its suitability for a given use, by this test as well as by other proposed tests, is questioned, since we cannot be assured that a steel, having very superior transition temperature and shear fractures, will not fracture suddenly and with so-called cleavage breaks under certain conditions of stress, design, and temperature.

In view of the findings in regard to transition temperatures as obtained by this test, since there is so small a difference in results for the more common grades of steel, we believe that some recognition should be given the suggestion of the author that those types of steel showing the best test results, such as steel for subzero-temperature service, which infers a fully deoxidized grain-size-controlled steel, should be used for construction of critically stressed portions of a ship or unfired pressure vessel.

W. BENNETT.⁷ The subject of cleavage or brittle fractures has been forcibly brought to our attention by a number of major and minor casualties in the case of some war-built welded ships.

The tests described in the paper show that the type of steel and the geometry of the notch are most important. For example, if the angle of the notch is large, the relative danger of a cleavage fracture is, as might be expected, less, and vice versa. The tests also appear to show that if the angle of the notch is small enough (approaching a crack), this condition will cause cleavage failure on plates regardless of type (composition).

While it is felt that such painstaking research as the author has evidently undertaken on this subject is undoubtedly of great value, the writer is of the opinion that the cause of these steel failures should not be laid wholly to the steel and that contributory causes were a combination of (a) excessive speed in construction, (b) improper welding sequence, (c) insufficient qualified supervision, and (d) inexperienced welders. These points are mentioned, bearing in mind that 60 "Ocean" vessels (similar to the "Liberty" type) were built early in the war years (1941-1942) at Portland, Me., and Richmond, Calif., under the supervision of Lloyd's Register (for the British Government), and on these 60 welded ships no major fractures developed. The steel was all of United States' manufacture, and was presumably the same average quality as used in subsequent war years for the Liberty ships and other types. This statement of fact regarding the service performance of these 60 Ocean-type vessels leads one to the conclusion that the problem is not simply one of poor-welding-quality steel. Brittleness is not something that is new to steel, nor is it something that has only been discovered since the recent war. It has been known for years, but obviously was not of such relative importance in the riveted as in the welded ship.

The author mentions the advantage to be derived by the use of a steel which would resist propagation of cleavage fractures, but does not give a definite opinion of the desired chemical analysis of such steel and the probable cost thereof.

In this connection mention might be made of the investigations

⁷ Principal Surveyor, Lloyd's Register of Shipping, New York, N. Y.

being made in the United Kingdom on this and related subjects. I see no reference to any of these in the Bibliography. Papers by W. Barr and C. F. Tipper,⁸ and by W. Barr and A. J. K. Honeyman⁹ would be of particular interest.

This experimental work was initiated at a conference held in Cambridge in October, 1945, at which it was decided to select an acceptance test to assess notch brittleness. It was decided to make a series of notched bend, notched tensile, and notched impact tests. During these tests Dr. Tipper developed a simple notch tensile specimen pulled in an ordinary tensile machine, which she claims simulates the same type of fracture as developed in some of the war-built Liberty ships and others.

The discussion on these papers indicates that impact is not necessary for the production of such brittle fractures, but that a notch is essential. It was also found that temperature is very important.

The synopses given by the authors on their respective papers are of sufficient importance to be quoted, as follows:

(a) "The temperature range of transition from tough to brittle fracture of mild-steel plates of different carbon and manganese contents was determined by means of notched-bar impact, notched bend, and notched tensile tests. The results obtained by each of these tests were in good agreement, except that for very soft steels the notched tensile test gave a lower transition range than the other two tests.

"It was found that the transition range is raised by an increase in the ferritic grain size, by an increase in plate thickness, and by slow cooling after normalizing. It was also found that a high notched bar impact value may be accompanied by a fracture which is mainly cleavage.

"Tentative conclusions, subject to confirmation, have been reached that the effects of plate thickness and slow rates of cooling in raising the transition range are reduced in mild-steel plates with higher manganese contents."

(b) "A series of four mild steels was made in which the only significant variable was the manganese/carbon ratio. The residual elements were low and the tensile strengths were approximately the same for each steel. The notched-bar impact properties of these steels in the annealed and in the normalized conditions have been determined. It was found that increasing the manganese/carbon ratio lowers the range of transition from tough to brittle fracture, increases the impact values at all temperatures, and tends to result in finer McQuaid-Ehn and ferritic grain sizes. A practical recommendation is made that for structural steels for shipbuilding purposes, the manganese/carbon ratio should be not less than 3-0."

What does the author think of the notched tensile test developed by Dr. Tipper and described in her contribution to the Cambridge Conference? Has he any experience with this form of test piece? It is suggested that sufficient information might possibly be obtained by this means at a lower cost in time and money, than by the employment of the special form of tear test referred to in the paper.

What would he think, too, of the suggestion that the manganese/carbon ratio be specified at not less than 3?

In conclusion it is suggested that the danger repeatedly demonstrated of the effect of a notch should be stressed; and as welded vessels are more liable to have "notches," visible and invisible,

in their construction, the importance of x-ray examination of welds and detailed inspection by experienced surveyors, is self-evident.

C. L. CLARK.¹⁰ It is gratifying that the author included standard impact tests in his program and found that the various steels were arranged in the same order in both tests. Many have derided the results from impact tests as they felt that the factors involved were seldom encountered in service. Yet these results indicate the impact test capable of revealing the susceptibility of steels to cleavage or brittle-type fractures, although it is not as sensitive in this respect as the author's test.

With respect to his work on welds, we would like to ask if any tests were conducted with the notch in the fusion zone, or in the gradient zone, rather than in the weld deposit proper, and if so, whether any differences were noted in the results obtained? Likewise, were welded specimens ever notched with the notch parallel to the fusion or gradient zone rather than perpendicular to the length of the specimen? It would seem that a test of this type would indicate the effect of lack of complete fusion on the type of fracture to be expected.

G. H. ENZIAN.¹¹ We have used the author's test, in a limited way, and believe it to have merit in explaining the failure of certain steels in such applications as welded ships, pressure vessels, and other structures subject to dynamic multiaxial stresses. However, as is the case with any new test, more data will be needed to establish conclusively its superiority over conventional acceptance tests, such as tensile and impact. Nevertheless, we are in full accord with his contention that tests, which employ the full thickness of the material actually used, are more reliable than carefully machined laboratory tests that, in many cases, remove certain portions of the steel which may have an important bearing on the reaction of the structure to service conditions.

We would like to comment, briefly, on that phase of the paper dealing with the "transition temperature."

It is generally recognized that a material behaves in a brittle or ductile manner according to such conditions of testing or service as temperature, rate of straining, and combined stresses. These three factors are believed to be interrelated, and therefore the behavior of a steel to changes in one of the factors can be used to predict its behavior on variation of the other factors. Thus a steel which is readily susceptible to embrittlement at low temperatures should also show a tendency for brittle behavior under the conditions of combined stresses imparted by the tear test or in actual service. However, this relationship is only qualitative and, as the author points out, "the state of stress determines the level of the transition temperature range and, unless the state of stress is defined, the transition range cannot be defined for a given metal."

It is quite possible that a better relationship than was shown in the paper between the transition temperatures for Charpy and tear-testing would have been obtained had the author used the same notches in both cases. It was our experience that when V-notch Charpy tests were compared with tear tests made with the same type notch (45 deg, 0.01 in. radius), the transition temperature for the Charpy bars actually was a few degrees higher than for the tear tests. Thus the transition temperatures with this type of Charpy test would have been more nearly similar to the transition temperatures found on monolithic structures in service, to which the author refers.

There seems to be little question that for certain critical sec-

⁸ "Brittle Fracture in Mild Steel Plates," by W. Barr and C. F. Tipper, *Journal of the Iron and Steel Institute*, vol. 157, October, 1947, pp. 223-238.

⁹ "Effect of the Manganese-Carbon Ratio on the Brittle Fracture of Mild Steel and Some Factors Affecting Notched-Bar Impact Properties of Mild Steel," by W. Barr and A. J. K. Honeyman, *Journal of the Iron and Steel Institute*, vol. 157, October, 1947, pp. 239-242.

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¹¹ Division Head, Research and Development Division, Jones & Laughlin Steel Corporation, Pittsburgh, Pa.

tions of a ship, a steel which is less notch-sensitive than the steels now used might be employed to advantage. However, of considerable importance from a practical view, is the question: "How much less?" This is admittedly a difficult question, but an acceptable answer might be deduced from statistical analysis of the frequency of ship failures, and by taking into consideration the number of ships which have been entirely satisfactory.

With the information available at present, laboratory tests, whether Charpy or tear tests, will show only the relative susceptibility to embrittlement of various materials and can be used to predict only which material is more likely to fail under given severity of service conditions. As yet, the limits within which the transition range should lie in order for the steel to be adequate for the most severe service conditions likely to be encountered, are not definitely known. It may be possible, however, that, since a large percentage of ships have apparently been satisfactory, a relatively slight improvement in the properties of the material may be sufficient. It would be extremely interesting therefore to have a comparison of tear-test results, along with other test data, on steels which failed in service and steels which were satisfactory under the same operating conditions.

FINN JONASSEN.¹² With reference to Table 11, and particularly to Fig. 15 of the paper, one wonders whether the author has succeeded in developing a test that is not subjected to the scatter usually present in the impact and "tear" test (1)¹³ when going from shear to cleavage fractures. Additional tests, particularly in the transition region, using the steels already investigated, and material from other heats will answer this question. In this connection, other thicknesses also will have to be investigated.

Recently completed work under the Ship Structure Committee (2) involved the static-testing of large centrally notched flat-plate specimens (3, 4, 5). It is generally accepted that these large specimens are representative of conditions approaching those obtaining in a real structure, e.g., a ship. These specimens have failed with both shear and cleavage fractures at "nominal" stresses (based upon maximum load and the original net area), varying from 30,000 to 45,000 psi for the wide specimens (72 in. wide), when tested at temperatures from -40 F to 125 F. Similar specimens, but narrower (12 in. wide) gave somewhat higher nominal stresses. These tests were all made on $\frac{3}{4}$ -in.-thick medium-steel plate of shipbuilding quality. It is of interest therefore to consider the possible reasons for the much lower stresses obtained on the cleavage-tear test for both shear and cleavage failures, as compared to the results obtained from symmetrically notched specimens on similar steels. Reasons for these differences are undoubtedly included among the following:

1 The eccentricity of the cleavage-tear specimen which affects the stress distribution in the specimen, producing, at the edge opposite the notch, low tensile stresses or possibly compressive stresses. The low tensile or compressive stresses obtained in this region of course will reduce the nominal stresses carried by the entire section.

2 The effects of cold-working the notch. This effect will serve to embrittle the steel adjacent to the notch through work-hardening and strain-aging. In this connection, the question is raised as to the length of elapsed time between the forming of the notch and the testing of the specimens. We know from related work that prestraining 10 per cent and aging can raise transition temperatures by as much as 80 deg F, as shown by the Charpy keyhole impact tests (1).

3 The effect of plate thickness. The related work involving

the symmetrical notch specimen was all done on $\frac{3}{4}$ -in. plate stock. The thicker section of the cleavage-tear tests should serve to increase the restraint of this specimen.

4 The effect of the sharper notch radius in the cleavage-tear specimens may have an effect on the transition temperature and fracture load.

It is of interest to note that the transition temperatures indicated by the tests of $\frac{3}{4}$ -in.-thick centrally notched flat-plate specimens (72 in. wide) are higher than those shown by the Charpy keyhole impact tests on the same steels, and that these differences range from 35 to 90 deg F with an average of about 67 deg F. The author also reports transition temperatures for the 1-in.-thick cleavage-tear specimens that are about 100 deg F higher than those obtained from the Charpy keyhole impact tests for the same steels in the "as-received" condition.

Attention is called to the author's findings when water-quenching killed and semikilled steels. Here a significant increase in the breaking load and particularly a lowering of the transition temperature by as much as 100 deg F has been achieved through the heat-treatment of medium steels.

This paper represents an important contribution to our ultimate understanding of the behavior of medium steel when subjected to triaxial stresses and low temperatures. The specimen developed will be of assistance in arriving at criteria for selecting steels best suited for safe welded construction.

BIBLIOGRAPHY

1 "The Correlation of Laboratory Tests With Full-Scale Ship-Plate Fracture Tests," by E. P. Klier, F. C. Wagner, and M. Gensamer, *The Welding Journal*, vol. 27, no. 2, Research Supplement, 1948, pp. 71-s-96-s.

"A Method of Evaluating Transition From Shear to Cleavage Failure in Ship Plate and Its Correlation With Large-Scale Plate Tests," by Noah A. Kahn and Emil A. Imbombo, *The Welding Journal*, vol. 27, no. 4, Research Supplement, 1948, pp. 169-s-185-s.

2 "The Work of the Ship Structure Committee," by Rear Admiral Ellis Reed-Hill, USCG, *The Welding Journal*, vol. 27, no. 2, Research Supplement, 1948, pp. 33-s-34-s.

3 "Causes of Cleavage Fracture in Ship Plate—Tests of Wide Notched Plates," by A. Boodberg, H. E. Davis, E. R. Parker, and G. E. Troxell, *The Welding Journal*, vol. 27, no. 4, Research Supplement, 1948, pp. 186-s-199-s.

4 "Cleavage Fracture in Ship Plate as Influenced by Size Effect," by W. M. Wilson, R. A. Hechtman, and W. H. Bruckner, *The Welding Journal*, vol. 27, no. 4, Research Supplement, 1948, pp. 200-s-208-s.

5 "A Study of Slotted Tensile Specimens for Evaluating the Toughness of Structural Steel," by H. R. Thomas and D. F. Windenburg, *The Welding Journal*, vol. 27, no. 4, Research Supplement, 1948, pp. 209-s-216-s.

N. A. KAHN¹⁴ AND E. A. IMBEMBO.¹⁵ From having studied the applicability of a tear test for evaluation of transition from shear to cleavage failure in ship plate, the writers find their results to correlate with the indications obtained from large-scale tests.

Our main interest has been directed toward the development of a test method which would be suitable for quality-control purposes. Such a method, in our opinion, would have to be based upon a test specimen which is economical with regard to material, time, and cost of preparation and testing, which of course is not the case with the test specimens used by the author. His test coupon, in some instances, was 30 in. $\times$ 30 in. and not less than 16 in. $\times$ 10 $\frac{1}{2}$ in. and involved complex prenotching preparation and reinforcing of the loading holes.

The test coupon developed in the laboratory with which the

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¹⁵ Senior Metallurgist, Material Laboratory, New York Naval Shipyard, Brooklyn, N. Y. Jun. ASME.

Note: The opinions contained in this discussion are the private ones of the writers and are not to be construed as reflecting the views of the Navy Department or the Naval Service at large.

¹² National Research Council, Washington, D. C.

¹³ Numbers in parentheses refer to the Bibliography at the end of the writer's discussion.

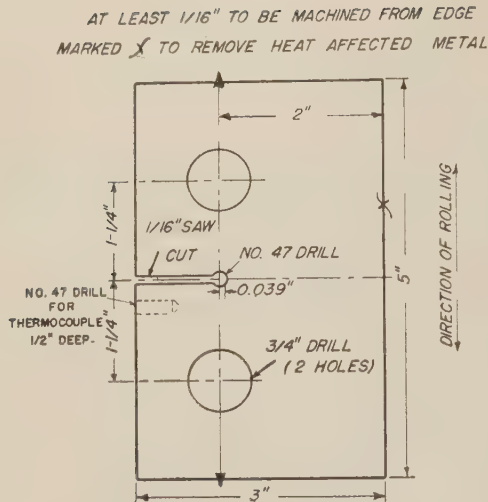


FIG. 22 TEAR-TEST PLATE SPECIMEN, KEYHOLE NOTCH



Tested at
65 F 50 F 35 F

FIG. 23 FRACTURES OF TEAR-TEST SPECIMENS CORRESPONDING TO DIAGRAMS OF FIG. 24

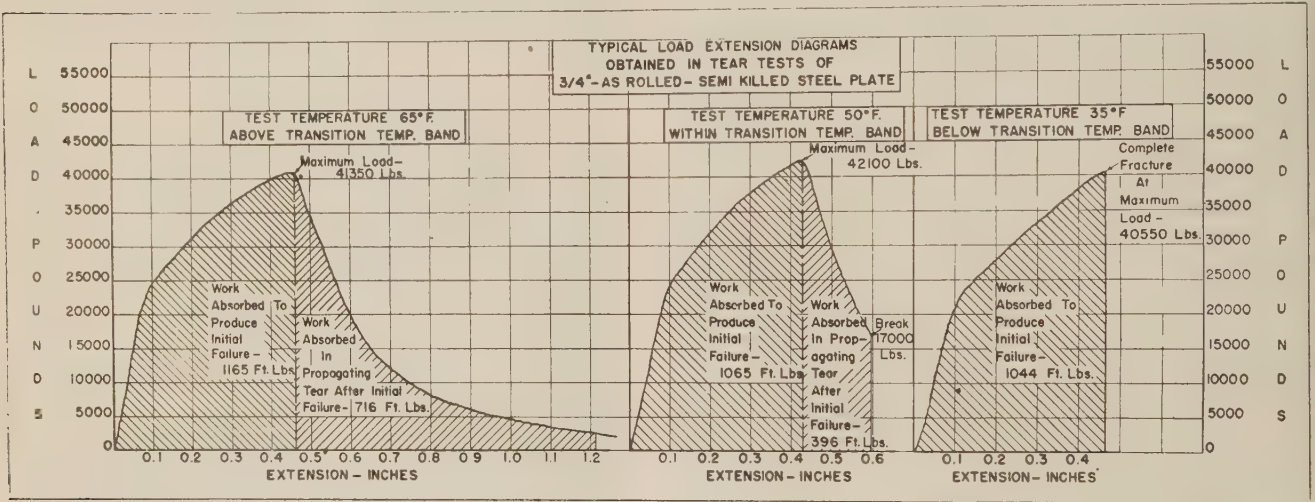


FIG. 24 TYPICAL LOAD-EXTENSION DIAGRAMS

writers are associated is shown in Fig. 22 herewith. The blank is flame-cut from full-thickness plate and machined only on the edge opposite the notch. The notch is of the keyhole type and is cut transversely to the direction of rolling. A load-extension diagram obtained with the test provides data for the evaluation of maximum load, deformation, energy absorbed in initiating failure (occurs at approximately maximum load), and finally energy input to propagate tearing from maximum load to completion.

Typical fractures of specimens tested at temperatures above, within, and below transition of a representative 3/4-in-thick ship plate are shown in Fig. 23. The corresponding load-extension diagrams are shown in Fig. 24 of this discussion.

It will be observed that the ductile fracture is associated with a high energy value to propagate tearing, the cleavage fracture with an energy-to-propagate value of practically zero, while the mixed fracture (first shear and then cleavage) is associated with an intermediate energy-to-propagate value which is roughly proportional to the percentage of shear in the fracture. It is to be noted that, in connection with the specimen used by the writers, the maximum load and energy absorbed in initiating tearing are relatively unaffected by the mode of fracture.

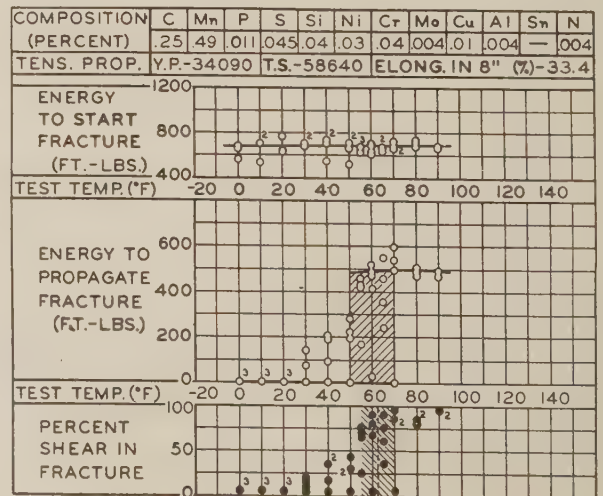


FIG. 25 TEAR-TEST PROPERTIES OF PROJECT STEEL "A" SEMI-KILLED, AS-ROLLED, 3/4-IN. PLATE

A summary graph of a tear-test series for a typical medium ship-plate steel, $\frac{3}{4}$ in. thick, is shown in Fig. 25, herewith. It is to be observed that the energy-to-propagate tearing, and the percentage of shear in the completed fracture sharply define the transition from ductile to brittle failure, and that the two indications practically coincide with each other. The foregoing appears to be in contradiction to the author's findings wherein maximum load or unit load defines the transition, while, in connection with the writers' findings, these are unaffected by mode of fracture. The disagreement may be due to the geometry of the specimen. We would be interested in having an evaluation of fracture work associated with the author's test coupons, if available.

We are in full agreement with the author that a tear test simulates several of the basic conditions of loading encountered in service, is well-suited for the study of conditions which will precipitate cleavage fracture, and facilitates the testing of full-thickness plate specimens with a minimum amount of machining work and with nominal testing-machine capacities.

The author points out that by proper design of the tear-test specimen, it is possible to have the entire section under tensile stress with a maximum at the notch and decreasing to a minimum at the edge opposite the notch. It is noted, however, that his recommended basic design for a tear-test specimen (coupon type D) with an eccentricity of loading of 4.59 in. and depth of 6 in. is subject to compressive stresses as well as to tensile stresses, since the neutral axis for this specimen, using Equation [6] of the paper falls 0.61 in. below the centroid. The fractures of the tear-test specimens used by the writers likewise show evidence of compression at the edge opposite the notch.

In connection with Table 8, the author states that notch radii smaller than 0.01 in. favor development of cleavage fractures, notches of the Izod (0.010 in. R) and Charpy (0.039 in. R) type will produce a combination of shear and cleavage fractures, while notches with radii greater than the Charpy will favor shear fractures. It is not clear from this what part the steel itself plays in determining the mode of fracture it is to follow under given conditions of loading temperature and degree of constraint.

The work, reported in the paper, on the effect of loading rate is interesting to the writers inasmuch as, for lack of adequate background, they have limited the rate of loading to 0.05 ipm. It would appear from Table 10 that rates of loading up to 2 ipm do not influence the unit breaking load. Since the breaking load defines the transition range in connection with the author's tear test, it would appear that the latter would likewise be unaffected.

A. B. KINZEL.¹⁶ The author is to be complimented on a keen analysis and an ingenious test specimen, as well as an elegant mathematical exposition which sheds much light on steel behavior in structures, particularly from the engineering point of view. The wealth of data given will do much to provide a better understanding and appreciation, by engineers, of the phenomena in question, especially with reference to mass effect. This is both the strength and the weakness of the particular approach herein reviewed. The specimen is not essentially different from that suggested by Coker many years ago and used for the same purpose. This use was not as successful as the author's. A few years ago our laboratories attempted a study of mass effect, using the C-specimen with the notch in the vertical leg of the C. We learned a lot, but as the simple C-specimen has shortcomings (many of which have been overcome in the present specimen), we did not believe that it was suitable for general application.

Some of the points mentioned by the author should be emphasized. Behavior of steel is affected not only by the complex stresses which he so neatly provides and analyzes, but also by the

rate of loading and temperature. In the work reported, the rate of loading has had no effect. This is probably due to the comparatively small differences in rates of loading used and, as he points out, the fact that the stress condition was so severe that the threshold level at which the rate of loading becomes highly important has already been exceeded. The data given on temperature likewise were not over a broad enough range to warrant conclusions, as practically all of the work was below the transition temperature. If, as the author assumes, we are to use steels which will show brittle behavior, this work below the transition temperature is highly important and, as far as we know, unique. However, the problem would seem to be to design structures and select steels so that in service brittle behavior is avoided. This is brought out by the author, and the writer is sure he will agree that, from the engineering standpoint, the properties below the transition temperature are of interest primarily in establishing transition temperature and engineering conditions which tend to raise or lower it.

Our own belief is that any test must include a complex stress such as that provided by the author's specimen, a rate of loading not too different from that in service (likewise provided by his specimen), and a major temperature effect of such magnitude that the test for brittleness can be carried out at a temperature commensurate with the lowest expected in service. The author's stress conditions are so severe that the transition temperatures are much too high to be representative of reasonably good design. This is borne out by experience with ships, where practically all of the failures took place at relatively low temperatures. The argument that all structures contain cracks and therefore super-severe stress conditions, when carried to a logical conclusion, indicates that most engineering structures made with the usual structural steels should fail from brittleness in service. This is not borne out by experience so we must examine further the effect of a notch.

The author brings out clearly that after failure has started in a few of his C-specimens, crack propagation ceases and there is contraction in dimension across the U. This is beautifully demonstrated in Figs. 18 (a) and 18 (b). This means that there is a condition at the root of a crack where the stress concentration due to that crack is insufficient to provide further failure. Argument from this point leads to the conclusion that the same condition would not have started the major crack if it existed at the root of a microcrack. Thus we believe that the importance of micro-cracks has been overemphasized.

We further believe that any welded test specimen should include to a major degree all heat-affected zones in the notched condition, this for general use, although we appreciate that some specific factors may be evidenced to greater advantage by special specimens, such as the author's.

There is one other point which should be emphasized in that it illustrates the difference in the point of view between engineers and metallurgists. The metallurgist differentiates clearly between strength properties and ductility or mode of failure. The engineer is interested in the stress condition which results in a given mode of failure, and thus expresses ductility in terms of fiber stress. It is our thought that in many structures in which there is constraint the local stresses automatically increase until motion for adjustment takes place. The question is: Does this motion take place by plastic deformation or by failure? In other words, in service the local stress will frequently be enough to provide motion, so that the important item is not fiber stress but behavior. As set forth in the 1947 Campbell Memorial Lecture of the ASM, we believe that this property can best be measured by observing the reduction of area, and we suggest that the author attempt a correlation of the reduction of area with this behavior performance rather than with unit stress as he has done.

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H. G. LAREW.¹⁷ In discussing this paper the writer will not endeavor to point out its many good features which are so obvious. Instead, two suggestions will be advanced for further study which may lead to results similar to those observed by the author but which are, however, more readily obtained.

It would be quite desirable to submit plastic models of the various specimen types to a series of photoelastic examinations. These photoelastic studies would show readily the extent of stress concentration which develops in the region of curvature at the bottom of the 4-in. slot. Results of these studies would probably show that the effect of this concentration of stress is not negligible at the notch. In that case it would be desirable to increase the 4-in-slot dimension until the notch area itself is free of all or most of this undesirable concentration of stress that develops in the region of curvature at the bottom of the slot.

Another possibility for further study is found in the development of Equation [5] of the paper

$$\frac{P}{A} = \frac{S}{1 + \frac{6ne}{d-h}}$$

In developing this equation, the author combined two classical formulas, one for axial tension or compression stress, and the other for flexural stress, into a single Equation [1]

$$S = \frac{P}{A} + \frac{Mc}{I}$$

This equation is not immediately applicable to the case of the cleavage-fracture specimen. The author therefore modified this expression so that the effect of stress concentration at the notch itself, and numerous other variables would be represented in the equation. This was accomplished by introducing a correction factor n , in Equation [3] and applying it to the Mc/I or flexural stress term only. It must be kept in mind, at this point, that the component of direct stress is also subject to the factor of stress concentration and the other variables mentioned by the author. The writer suggests that a similar correction factor called m , be applied to the P/A term. If this additional factor m , is introduced, Equation [5] will become

$$\frac{P}{A} = \frac{S}{m + \frac{6ne}{d-h}}$$

This new factor m , is variable and will be affected for the most part by changes in the values of d and R . It will increase in value quite rapidly as the ratio of R/d decreases.

The fact that a correction was not applied to the P/A term of Equation [3] may explain why the author found that Equation [5] did not give satisfactory results when the depth of the specimen was increased appreciably. The value of m will increase with increasing specimen depth. If it were applied to Equation [3], the third term in the denominator of Equation [10] would not be needed. This should simplify Equation [10] and the calculations considerably. However, it may not change appreciably the over-all results observed by the author.

E. M. MACCUTCHEON.¹⁸ This paper describes a very complete and well-organized research project. Working independently, the author has paralleled a large portion of the research carried on by the board investigating the ship failures and now

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¹⁸ Naval Architect, Navy Department, David Taylor Model Basin, Washington, D. C.

under the surveillance of the Ship Structure Committee. His mathematical approach in the presentation of the results is a challenge to the designers who must ultimately interpret such data. This phase of the paper was most interesting to the writer who will discuss it in detail after presenting a few remarks on crack arrestors.

The author states that the objective of the riveted crack arrestor was to obtain structural flexibility as well as to arrest cracks. As one of the three original sponsors of this structural modification, the writer has taken every opportunity to deny this widespread impression. The riveted crack arrestor was intended only as a crack arrester. Flexibility was not a virtue claimed in the original proposal. The writer is not sure that he knows what is meant by the term "flexibility," as used in this paper in relation to ship structure. This term has been bandied about and used loosely to express several different things.

With regard to the radii of holes intended as crack arresters, it should be pointed out that this is a quantitative and not a qualitative question. The tests indicate a definite trend, but only statistics from actual failures can establish the base line. The tests indicate that the larger the radius of the hole, the better its crack-stopping ability. Obviously, the optimum hole is one of infinite radius, i.e., a slot.

The following statistics from the records of ship performance may be interesting in view of the unequivocal statement in this paper that it is simply necessary to drill a hole in order to stop a crack:

	Crack passed through	Crack stopped
Rivet hole.....	5	13
Slot.....	0	13
Crack arresters as a whole..	0	33

The foregoing data are based upon structural failures only. Crack arresters were also reported effective in many cases of secondary damage initiated by grounding or by mines and torpedoes. The decision to incorporate a slot in the arrester appears to have been justified.

The author's proposal to use perforated straps without slots is not entirely new. The first proposal for remedies to the Liberty ships included riveted straps with no slots. Records of some failures indicating the inadequacy of such arresters were already available at the time of the final decision. An example was the steamship *Champlain*, a Great Lakes ore carrier. On this ship, cracks passed right by four riveted longitudinals, and in one case, right through a rivet hole. They did not crack the longitudinals but they did not stop either.

The edges of straps should not be welded as suggested in the paper. Tests performed by the Navy Department about 4 years ago showed that light welds between two heavy plates can serve as a bridge across which a crack can jump from one heavy plate to another. Furthermore, if such a plate were attached without plugging the holes with rivets, it would have the same practical disadvantages with regard to leakage between faying surfaces which are present in the crack arresters now in use.

On reviewing the computations and curves plotted by the author, which show the load P on the specimen divided by the net gross section area A , the writer was startled by the figures indicated. At first glance, he misinterpreted these P/A values as being actual nominal stresses, without appreciating the effects of the eccentricity involved. Upon further reflection and upon consideration of other tests, he began to suspect that the stress-concentration factor of 9 cited in this paper might be misleading, and that further definition of the parameters involved in the test should be made. One other statement, to the effect that the yield point was increased almost to the breaking strength, also appeared misleading.

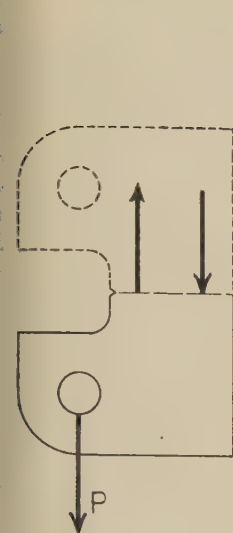


Fig. 26

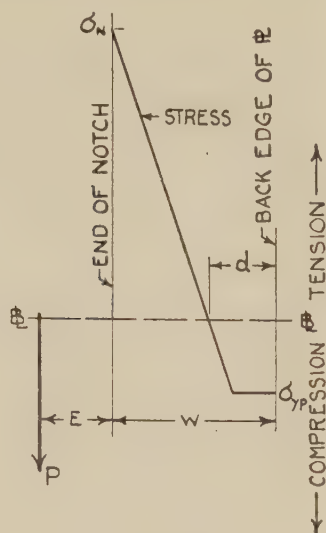


Fig. 27 CASE I

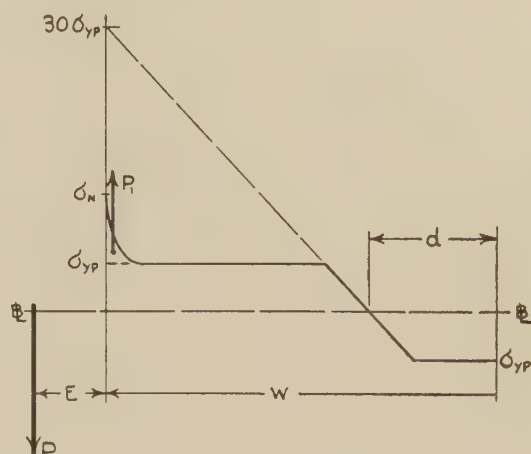


Fig. 28 CASE II

To clarify his own thoughts on this matter and in the hope of putting his mind at ease, the writer attempted to rationalize the results mathematically. In doing this, the necessary balances between forces and between moments were established (see Fig. 26 herewith). In making the calculations, only the tensile-bar data for the particular steel, steel 17, were used, and an effort was made to compare the results with the notched-plate results presented in the paper.

In preparing the calculations, the following assumptions were made:

- That the failure occurs in the cleavage mode.
- That the degree of brittleness of the cleavage failure corresponded to shipboard failures. The reduction in thickness was assumed to be 2 per cent near the starting point of the fracture. This would correspond to 1-1.5 per cent where the fracture is running, and this thickness reduction is typical of the ship failures examined at the National Bureau of Standards.
- That the material has a pronounced yield point.
- That the stress distribution across the section opposite the notch corresponds to Figs. 27 and 28 of this discussion. Case I is for narrow specimens of width on the order of the plate thickness. In such specimens, the opposite edge affects the stress distribution around the notch. Case II is for wider specimens and the intercept of the elastic line in Fig. 28 was determined by the assumption of 2 per cent thickness reduction at the notch as just mentioned.

In Case II, the load at the end of the notch indicated as P_1 is an unknown from the data on the tensile-test specimens. Data from a previous report¹⁹ were used to evaluate P_1 . Iso-hardness lines around the notch were converted to strain and then to stress by the calibration curves provided in the reference just mentioned, for the cleavage failure of a 72-in. centrally notched specimen of steel C. Making due allowance for the difference in material characteristics between steel C of the reference¹⁹ and steel 17 in thickness of 1 in. of the present paper, the value of P_1 was found to be about 20,000 lb. Equating the forces and moments for the stress curve for Case I, shown in Fig. 27 of this discussion, gave the following equations

$$d^2 + d \cdot 2 \cdot \left[(w - d) + E - \frac{0.01 \times t \times D}{w - d} \right] - (w - d)^2 \left[\frac{(\sigma_N + \sigma_{YP})^2}{3 \sigma_{YP} \cdot \sigma_N^2} - 1 \right] - (w - d) \left(\frac{\sigma_N^2 + \sigma_{YP}^2}{\sigma_N \cdot \sigma_{YP}} \right) \left(E - \frac{0.01 \times t \times D}{w - d} \right) = 0 \quad [18]$$

$$\frac{P}{A} = \frac{w - d}{w} \cdot \frac{1}{2} \cdot \left(\frac{\sigma_N^2 + \sigma_{YP}^2}{\sigma_N} \right) - \sigma_{YP} \cdot \frac{d}{w} \quad [19]$$

In similar manner, the stress curve for Case II, shown in Fig. 28, gave equations as follows

$$d^2 = \frac{(w - 2d)^2}{2} + (w - 2d) \cdot E + \frac{P_1}{\sigma_{YP}} \left(\frac{E + 0.3 \times t}{t} \right) \quad [20]$$

$$\frac{P}{A} = \sigma_{YP} \frac{(w - 2d)}{w} + \frac{P_1}{w \cdot t} \quad [21]$$

where, in the foregoing equations

t = plate thickness

D = distance between pulling holes

For P , A , w , d , E , σ_N , σ_{YP} , and P_1 refer to Figs. 27 and 28, and text.

Values for comparison with the test were then obtained by substituting into the equations the following material data for steel 17:

$$\begin{aligned} \sigma_{YP} &= 32,900 \text{ psi} \\ \sigma_R &= 133,000 \text{ psi} \\ \sigma_{Y'} &= 127,700 \text{ psi} \\ P_1 &= 20,000 \text{ lb} \end{aligned}$$

To simplify the mathematics, the equations were formulated on the basis of the initial thickness of the specimen. At the time of inception of failure, the thickness reduction at the notch was assumed to be 2 per cent. For this reason, the actual rupture strength was adjusted and the value σ_N was used instead.

The results of substituting these values in Equations [18], [19], [20], [21] are shown as a solid line in Fig. 29 herewith. Corresponding test results taken from Table 4 of the paper are indicated as circles. Evidently P_1 was underestimated, because the theoretical

¹⁹ Cleavage Fracture of Ship Plate as Influenced by Design and Metallurgical Factors (NS-336) Part II—Flat Plate Tests," by H. E. Davis, G. E. Troxell, E. R. Parker, and M. P. O'Brien, OSRD No. 3452, Jan. 10, 1946.

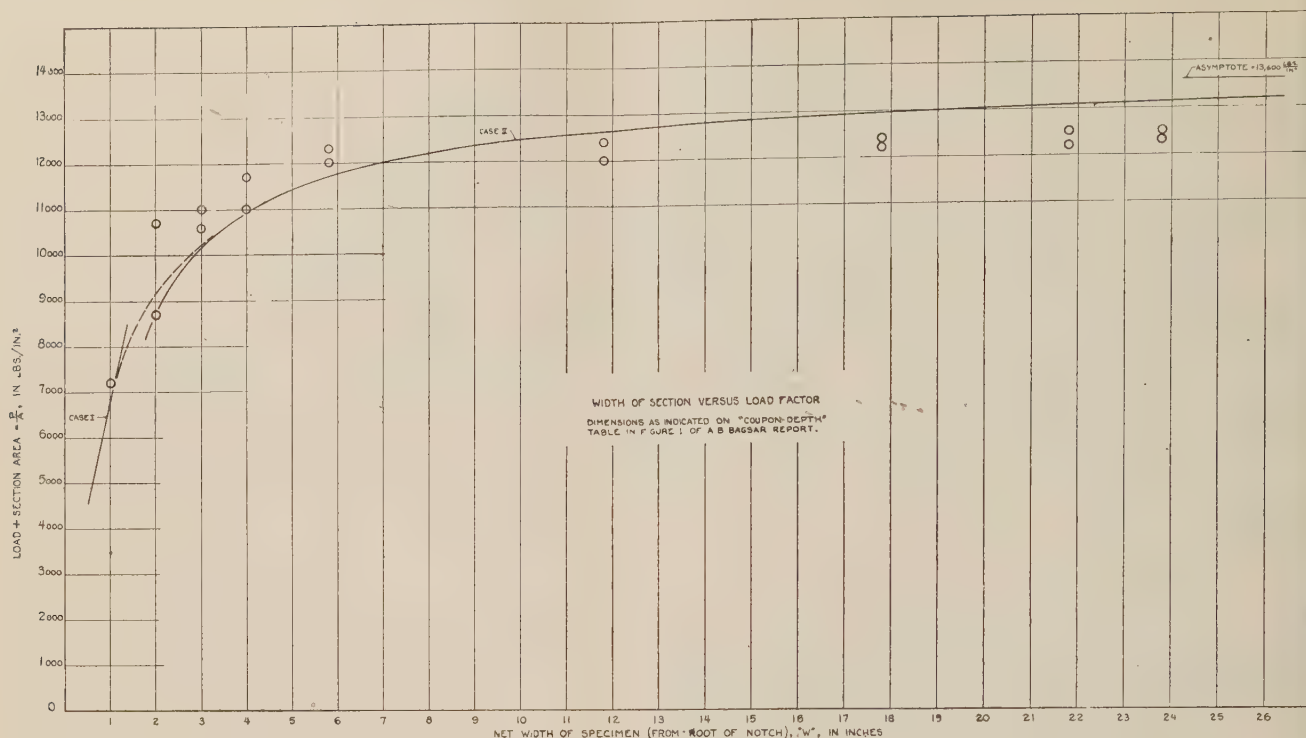


FIG. 29 WIDTH OF SECTION VERSUS LOAD FACTORS
(Dimensions as indicated on "coupon-depth" table in author's paper.)

cal curve lies below the test points for specimen widths less than about 8 in. It is probable that the degree of brittleness in the author's tests on steel 17 was not as great as that in the 72-in. plates of steel C at 32 F, on which the Knoop hardness values were recorded in the previous reference,¹⁹ and upon which the estimate of P_1 was based. The theoretical curve lies above the test points for wide specimens because buckling occurs. On the 22-in-wide specimen, the compressive stresses at the top and bottom edge of the specimen were estimated to be in the order of 20,000 psi. On the edge of the specimen opposite the notch, it has already been indicated that the compressive stress exceeds the yield point. The critical buckling stress for the loading indicated is probably not much over 15,000 psi and buckling is almost certain to have occurred. Perhaps the author can confirm this by his personal observations of the test.

The mathematical analysis used in producing the curves in Fig. 29 of this discussion was also applied to the centrally notched plates of steel C which were tested in widths varying from 3 to 108 in. and recorded elsewhere.²⁰ Good agreement between theoretical and test results was found again in this comparison for the centrally notched specimens. The type of loading on the centrally notched plates is more like that in a ship's deck. It is interesting to know that the results of both test and calculations indicate that the P/A value is asymptotic to the yield stress of the material.

The wide centrally notched plates cannot be compared directly to the ship's hull, however, because in considering the effect of small notches, it must be remembered that gross operators, such as hatch corner reinforcements, raise the stress values locally over a fairly large region. Plastic flow around a small notch might not reduce such gross stress concentrations appreciably. Fail-

ures could then be expected at a nominal stress in the deck somewhat below the yield point.

It appears that P/A values may serve as a useful tool even where notch sensitivity is involved. If credit is given for notch effects, designers may still be able to predict actual structural performance intelligently using P/A values. In a test for steel, P/A values for eccentric specimens are relatively insensitive to changes in the fundamental steel characteristics. However, it would appear from Fig. 15 of the paper that the P/A values may still serve a useful purpose as a means of discriminating between steels on a basis of notch sensitivity.

The mathematical approach to notch problems suggested by the author certainly captures the imagination. It presents intriguing possibilities for the designer but considerably more experience is needed in this field. We should be cautious in attacking existing strength standards. Such standards have proved generally satisfactory over many years. An apparent conflict between new theories and the standards may be an inadequacy in the theory rather than in the standards. Structural failures in ships' hulls are not the rule. They are the exception to the rule. More profit would be forthcoming if the theory could be made to explain these departures from the norm than by attacking the norm itself.

W. P. ROOP.²¹ The paper is most timely and constitutes an important contribution to the subject, distinguished from other work in the field by the originality of the test method used, and by the considerations advanced for relating the results of this test with others hitherto more widely known.

The number of parameters introduced and the ranges of their variations are so great that complete coverage was not attempted. In this respect there was no departure from the conditions of other tests. The main novelty lies in the form of the specimen and the test procedure.

²⁰ "Causes of Cleavage Fracture in Ship Plate, Flat Plate Tests and Additional Tests on Large Tubes," by H. E. Davis, G. E. Troxell, E. R. Parker, A. Boodberg, and M. P. O'Brien, Navy Contract NObs-31222, Jan. 17, 1947.

²¹ Swarthmore College, Swarthmore, Pa.

The expression "cleavage-tear" refers to two of the main features of the test, namely, that the rupture is accomplished by a tearing process, and the tests are mostly made at a temperature low enough so that cleavage predominates. Another special feature is that the main item of response of the specimen to the load which is made the subject of observation is the load itself, either the ultimate value or the value at fracture.

The load is tensile, but the eccentricity of the specimen is such that bending predominates. In this respect the test resembles that developed at the New York Naval Shipyard by Noah Kahn and associates. The eccentricity is generally even greater than in the Kahn test. However, provision against strain on the compression side, as by the hard pin in the Schnadt-Gensamer test, is made only by using specimens of great "coupon depth," d . Details of the specimen are shown in Fig. 1 of the paper; as a result of varying the dimension d a standard value of 6 in. was adopted, as compared with 2 in. in the Kahn specimen.

In the conduct of the test the later stages of the fracture are generally disregarded and the only quantity noted is the value of the maximum load. The decision to do this was reached as a result of a small number of tests made on a specimen of simple beam form, in which it was noted that high deflections at fracture were accompanied by relatively high loads at fracture. This, and other more general considerations are believed to validate the use of load as a discriminant of toughness, since load is correlated with deformation and hence with energy absorption.

The condition which must be satisfied to make breaking load a valid discriminant is said to be that "the proper size and type of coupons" be used. "It was necessary to use certain standard types of coupon and test conditions for determining the effect of a single variable in each series of tests."

The major part of the paper is devoted to finding the effect on breaking load in cleavage mode of the following parameters: (1) eccentricity; (2) coupon depth, i.e., depth of metal behind the notch; (3) depth of the notch; (4) angle of the notch; (5) radius of the notch.

The basic assumption underlying the method used is that the presence of the notch affects only that part of the stress which proceeds from the eccentric action. Thus the equation

$$\text{Stress} = P/A + (Mc/I)$$

gives elastic stress under eccentric load at distance c from the centroid in an unnotched specimen. This is assumed to become

$$\text{Stress} = P/A + n \cdot (Mc/I)$$

in the presence of a notch.

At a temperature far enough below the transition the action is considered to be elastic all the way to failure; the case of large plastic flow followed by cleavage fracture is stated not to occur in this type of specimen. Under these conditions the stress at fracture is called the cleavage strength. This usage is arrived at by the following process of analysis:

In the first series of tests described, unit breaking load is observed as a function of eccentricity. The data obtained in this way are given in Table 4 of the paper. Take, for example, the case of load parallel (fracture transverse) to the direction of rolling, notch angle 60 deg, coupon depth, and notch depth and radius not stated. Data cited are from 16 specimens, of 3 steels in 3 thicknesses. Since the scatter seems small, averages are taken to obtain 8 values of load at fracture on eccentricities running from 0 to 13 in. The plotted data, shown in Fig. 6 of the paper, indicate constant values of S and n as inferred from the observed values of P/A , e , d , and h by the equation

$$P/A = \frac{S}{1 + n \cdot 6e/(d - h)}$$

Turning to the data in Table 5 on effect of coupon depth, the same values of S and n as in Table 4 are used in reducing them, but a corrective term is introduced in the denominator. When depth exceeds 6 in. further increase has little effect; the value of 6 in. is a threshold. This is described by the author as the limiting depth "required to obtain the required rigidity and to compensate for the stress gradients created by the notch."

From a somewhat different point of view, it might be said that the 4-in. opening between the pins acts as a notch. Increased coupon depth is equivalent to reduced relative depth of this notch. At all depths over 6 in. the prongs loaded by the pins are in effect cantilevers, but at smaller depths the metal joining these prongs shares in the flexure.

As used by the author, however, the word "notch" refers to the small cut or indentation, not over $1/4$ in. deep, in the edge under greatest tension. The relative depth of this notch was thus very small at most. Yet Fig. 9 of the paper shows that the depth of this notch has very great effect on breaking load.

The suggestion that by making the notch depth small enough the cleavage strength can be raised above the shear-strength value, is one that would be of interest to confirm experimentally.

The effect of notch depth is described numerically by the correction factor in the numerator shown in Equation [11], found by a process of curve-fitting. The whole numerator, including the factor, is regarded as the cleavage strength. The value given in Equation [9] is understood to be that for a notch of $1/32$ in. or more in depth beyond which the change is negligible. As depth is reduced below that point, breaking load rises abruptly. According to Equation [10], as h approaches zero, the breaking load rises toward an infinite value. Before getting that far, no doubt, the fracture presumably becomes ductile, and the specimen behaves more as though the notch were absent.

A similar curve-fitting procedure is used for reducing the data on notch-angle, and the correction is again a factor in the numerator. The factor, and hence the cleavage breaking load, rises as the angle diminishes. For some reason the combined product of the three terms in the numerator is not given in Table 7. Would this again, as in Table 6 of the paper, be called cleavage strength? At what value of notch angle would it rise to the shear-strength value?

The answer to these questions is implicit in the result obtained for the next parameter, notch radius. This is accounted for by a third corrective factor in the numerator in which radius is raised to a fractional power. The whole numerator takes a value of about 200,000 psi, of the same order as the "technical cohesive strength," when the radius factor is introduced as a negative fractional power, and the value of radius is taken to be 3×10^{-6} in., the smallest value found by microscopic observation of natural cracks. Equation [15] is to be understood as giving the load value, taken in proportion to that for a standard notch. It would seem to be a little clearer if instead of $2.085 \times R^{0.113}$ the factor were written in the form $(R/0.0015)^{0.115}$. However, if the numerator is to rise to the value of 200,000 psi, the standard radius must be taken as 3×10^{-6} in. instead of 0.0015 in.

Table 8 of the paper deserves special attention as the first in which the temperature of the test is stated. The table shows a shift from cleavage to shear as radius increases. This would seem to confirm the idea that small radius favors high transition temperature. The data quoted, however, show a variation of breaking load with notch radius which covers a range of the latter quantity extending to between 4 and 5 powers of 10. The effect on load is confined to increase by a factor of 3. This jump of a mere threefold value of load when radius is multiplied by 30,000 might be simply a matter of transition from a lower to a higher level. The conclusion reached by the author, however, is that the shift follows a regular progression as expressed by the factor $R^{0.113}$, and

presumably for higher values of radius still larger values of breaking load would be found. However, as the author later says, "more data are needed to determine whether or not a leveling-off occurs at some value of notch root radius."

In the tests reported in Table 11, five series of observations were made under conditions that were uniform for each series except for a methodical shift in temperature. Although a transition is found, below which a lower level of breaking load occurs, the loss is not over about 25 per cent. No attempt was made to express this effect in numerical terms.

The transition thus found lay up to 100 deg higher than that found by keyhole Charpy tests, confirming Kahn's results.

In several respects the results of these tests appear to differ from others of which examples could readily be cited. These divergences may be attributed in part to the form of the specimen, but more especially to the fact that the primary quantity observed is a load in cleavage mode, and not a deformation or energy value. Examples are as follows:

1 In Table 6 no systematic effect of thickness is discernible. In other tests metal in great thicknesses has often shown reduced strength and toughness.

2 The author states, "It is obvious that no appreciable yielding is possible in fractures of strictly cleavage type." With other specimens, numerous examples have been found of fractures having all other marks of cleavage, but preceded by plastic elongations of several per cent.

3 In Figs. 8, 9, 10, and 11, the fitted curves pass through the points with very little scatter and dispersion. Upon repetition under identical conditions of tests, involving possible alternatives between two different regimes, such as the brittle and ductile modes of rupture, scatter is normally quite large.

4 Again, the author states, "Breaking loads are not affected materially by the shallow cold-working found at the base of the pressed notches." Cold work, as around a punched hole, has in some cases made the difference between a test which does and one which does not discriminate between similar steels differing in notch toughness.

5 Further "...cracks if present are very dangerous since relatively a much smaller load will be required to extend them in the sound structure..." In flat-plate specimens, cleavage cracks often stop by themselves and require increased unit load on the remaining section before proceeding further. The continuation is in some cases in ductile mode.

6 In conclusion 15, "...our present factors of safety are inadequate to safeguard against possible failures originating from notches or discontinuities." Some of the cleavage fractures in service occurred under such low load values that a moderately increased safety factor could hardly have served to prevent them.

These statements and counterstatements are seen to be not wholly contradictory although they do present opposed aspects of the matter. To arrive at statements to which no exception at all can be taken is a matter of such difficulty and would require so much more information than is now at hand that it will not soon be done.

It is the merit of this suggestive and stimulating paper that it brings to the subject a novel approach, presented in detail and without dissimulation, in the proving of which we would be sure to come nearer to our goal.

R. E. PETERSON.²² Since our company operates a number of outdoor pressure vessels of welded type, we are naturally concerned about winter operation and therefore are interested in findings of the type discussed in this paper.

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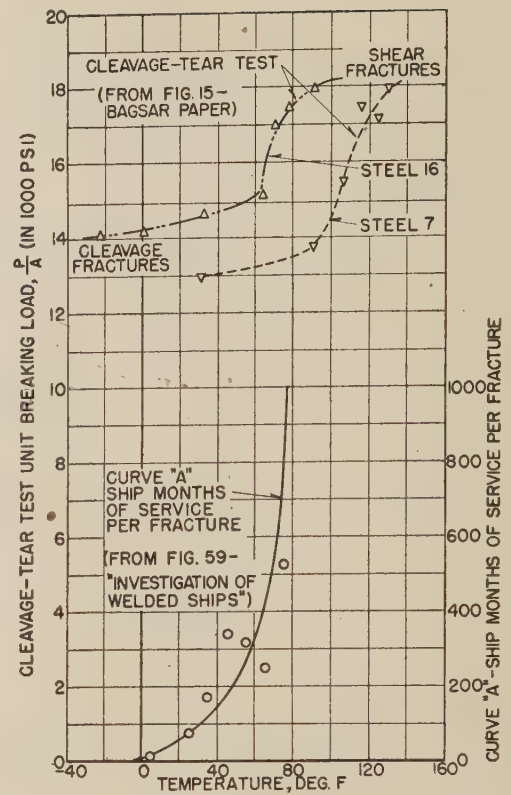


FIG. 30 CLEAVAGE-TEAR TEST
(From Fig. 15, author's paper.)

In Fig. 30 of this discussion, the writer has reproduced only those curves from Fig. 15 of the paper which show a definite lower branch; a curve of ship failures vs. temperature is also shown.²³ From the author's curves, it would seem that not much difference would be expected over the range -40 F up to 40 F or even 60 F. As shown in the lower curve, this is contrary to the experience with welded ships.

In this connection the writer wonders to what extent thermal stresses might explain the difference. In case of a drop in air temperature, the portion of the ship below the water and also the entire interior of the ship would remain warm, so that the condition would be like "quenching" the skin of the part of the ship above the water, with the possibility of quite high thermal stresses. These stresses superimposed on other stresses might help to explain the increased tendency for welded ships to fail at low temperature.

Since most outdoor pressure vessels would be in a more favorable condition is so far as such thermal stresses are concerned, the writer would appreciate the author's opinions regarding the views stated.

WALTER SAMANS.²⁴ In connection with wide interest in design, fabrication, and service difficulties experienced with unfired pressure vessels of all types of construction and sizes, it was a pleasure to note such a thorough analysis of the type of test specimen prepared by the author after painstaking work covering several years, which would evaluate the behavior of steel plates as rolled when a notch is present.

²³ "The Investigation of the Design and Methods of Construction of Welded Steel Merchant Vessels" (Government Printing Office), Fig. 59.

²⁴ Administrative Engineer, Sun Oil Company, Philadelphia, Pa. Mem. ASME.

The principal purposes of the tests, as stated, were the study of fractures that occurred in numerous welded ships. As most of these fractures were started by notches imposed by the shape, such as sharp hatchway-corners, similar details at gunwales, or in the extended strake forming the bulwark, it is expected that the specimens would illustrate that type of arrangement.

A similar condition exists in details on pressure vessels around openings, particularly where the reinforcement is unsymmetrical in relation to the face of the shell plate, and in such attached parts, usually outside of the vessel, which form the supporting lugs, brackets, or similar items that carry external loads, such as piping, or superimposed other vessels, heat-transfer equipment, etc.

Consequently* these tests are of great value in that they point out the unsatisfactory choice of Charpy impact tests as a gage for classifying steel.

In large atmospheric-temperature storage vessels it must be obvious that mild-carbon steels may be subjected to conditions for which the steel may not be suitable, unless it is carefully streamlined at critical details, even when subject to permissible design stresses under existing pressure-vessel codes.

Assuming, however, that the results of the author's tests are to be used in revising the present rules for pressure vessels, his suggestion that lower design stresses be set, obviously is meant to be applied only at the critical details mentioned and not to the entire vessel.

In this respect, some changes have already been made within the last 2 years, as for instance in the better material for attachments to supporting columns for large spheres containing liquids. It is also now required that the assemblies of parts be welded together in the shop and stress-relieved. Again, it is on these large vessels where the economy of proportions, in number of columns compared to the load to be carried, requires large details which are bound to have points of high stress concentration.

The mathematical treatment of the results obtained from the author's tests needs no apology, because he frankly states that additional data are required before more than empirical factors can be added to the basic formula in which the bending stress is added to the direct stress as is customary in all similar problems. While this neglects the elastic theories to some extent, the use of such theories at this time would not improve the results or conclusions reached.

However, there have been types of failures in pressure vessels which indicate forms of notches not covered in the present test. The writer has in mind one welded sphere under the hammer test when filled with water, and constructed by a well-known builder for an Army Ordnance operation, in which a hammer blow at ordinary temperatures, believed not to be lower than 40 F, caused a cleavage crack to be initiated at a defect in a butt weld along a median line at some distance above the equator of the sphere.

It is needless to say that any statements made at this time, based on experience and tests to date, are always subject to revision, and that if our program for research on unfired pressure vessels can be carried through on a number of construction details not yet covered to date, we may have a much better conception of how a pressure vessel should be designed for maximum safety and minimum cost of material and fabrication.

It is hoped that other investigators will use the data submitted by the author and apply them to tests of special steels of all kinds, good and bad, which may be available, because only by such means can we determine the boundaries of safe use and the hazards that should be avoided.

J. W. STEWART.²⁵ The author suggests consideration of pres-

²⁵ Director, Sir Joseph I. Sherwood & Company, Ltd., New York, N. Y.

ent factors of safety against large cleavage fracture due from small discontinuities, notches or cracks, or other remedies minimizing damages from cleavage fractures.

The paper endeavors to show that these cleavage fractures conform to some laws and modified equations, and that such fractures can be evaluated with reasonable accuracy. He endeavors to indicate certain design and material specifications by which occurrences of cleavage fracture can be minimized (see Fig. 1 of the paper). These are reduced to formula as follows

$$\frac{P}{A} = \frac{S}{1 + n \frac{6e}{(d-h)}}$$

Table 1 shows the test relationship on various notches. In this investigation eight steels were used from 1/2 in. to 2 1/4 in. thick, then butt-welded; seams in 1-in-thick plates welded by the manual and submerged-arc processes, the latter being accepted for the tests.

The numerical values of Equations [7], [8], and [9] show interesting results in regard to the correction factor.

Underbead cracks, undercuts, or discontinuities in parent metal, causing irregularities, constitute notches from which fractures occur, and show how mild steel may fail, by shear fractures, or both, depending on conditions. These are the extent of restraint to the flow of the metal in a direction transverse to the fractures and the temperature prevailing. It is noted that cleavage welds appears "sugary" or "crystalline." It has been stated that the rate of propagation of cleavage fractures is 3500 to 4500 fps (rapid rate). This is interesting and has been demonstrated by the ruptures that have occurred in the past, and recently on the tanker *Ponagansett*. This rupture was similar to that on the steamship *Schenectady*, the latter with only 9900 lb tension on the deck.

It is also noted that shear fracture has a "silky" appearance with slow propagation, and the extension of the fracture is gradual, differing in cleavage separation of the metal along cleavage planes, Fig. 17 (views 1 and 4); whereas shear is of the nature of sliding of one part over another, as in Fig. 17 (views 2 and 3). It is also interesting to note in the case of the cleavage sample, Fig. 18 (a), that 2 1/2 in. below, the notch was arrested by interposing shear fracture, finally resulting in a narrow band V-shaped fracture acting as a barrier against the cleavage; consequently this delayed rupture.

The V-herringboning appearance points to the origin of the crack in cleavage and opposite in shear fracture, Fig. 19 (b).

Table 8 shows the effect of V-notches and drilled holes when shear takes place and is less dangerous as it is always associated with deformation. Whereas in cleavage fractures there is no deformation, hence with no appreciable amount of energy absorption, so that the fracture appears suddenly, as soon as a certain load level is reached. This also has been indicated in recent ruptures. This apparently can be overcome or minimized only by either/or both modification of design or selection of better materials such as inserting riveted seams to break up monolithic structures.

The writer states that to arrest cleavage it is necessary to increase the notch arrangement and notch radius. This can be done by drilling holes of certain sizes so that when cleavage fractures develop they will terminate at one of these holes. The author states that the holes require no rivets except for watertightness. He further believes that to obtain watertightness it is only necessary to fit butt straps over the perforated area, either externally or internally, or both. These are made tight only to prevent leakage. Any reinforcement in addition are for the section weakened by the holes.

If desired, the holes could be drilled in the butt straps only and

these installed by welding at the desired section and in the desired direction. It is believed that the holes drilled in the butt straps will have the same effect for arresting cleavage as those drilled through the shell plate.

The writer is in disagreement with this and believes it advisable to fit a riveted seam. Experiments, the author states, show that holes need not be larger in diameter than $\frac{1}{2}$ the thickness of the shell plating (see Figs. 21a, b, c, and d of the paper). The writer believes it to be unnecessary to cut the shell or to relieve locked-up stresses. It is advisable to cut the shell and stop the cleavage propagation.

Specially deoxidized or denitrated steels and fine-grained structure would undoubtedly be advisable for certain critical application and might be used in modification, but they are expensive and practically unobtainable.

Therefore, for hulls of vessels it would seem desirable to continue a practice that has proved a safeguard, namely, to sever the vessel at the highly stressed sections and fit overlapping riveted seams, or riveted butt straps, for example, on the deck and on the bottom of tankers. Do this instead of producing steel of the proper coupon raised to coincide with the breaking load of the steel to suit notch and testing sensitivity. Otherwise, use a greater factor of safety which of course would mean increasing the weight of the vessel material through the main body for an all-welded ship as compared with a riveted vessel.

In his conclusions the author indicates the value of stress-relieving, but this is almost impossible in a structure such as a ship. Even normalizing is very expensive as mentioned in conclusion 11. However, a good technique of depositing welds in the correct sequence, as described in a paper by W. A. Stewart,²⁶ would help relieve locked-up stresses which have caused a lot of our ruptures.

In conclusion 12, on transition of temperature at 100 F, the writer believes this might be accepted in view of ruptures occurring in cold weather.

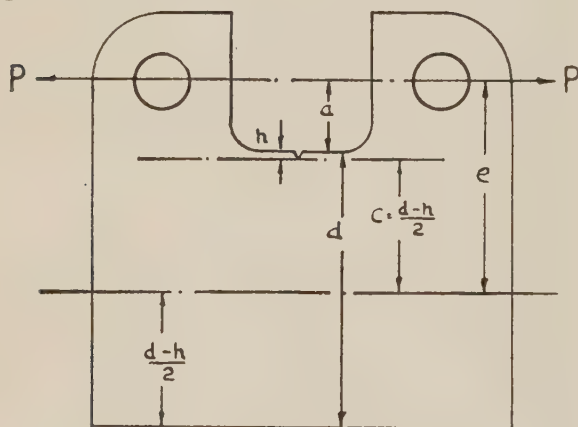


FIG. 31

The writer believes, and the practice so indicates, that to avoid major ruptures in a ship, the fitting of about four riveted connections, two topside and two on the bottom, together with good welding sequences, is the simplest answer to our present trouble in an all-welded ship structure.

J. GIBSON²⁷ AND C. F. TIPPER.²⁷ There appear to be some discrepancies between results obtained in this paper and those of other investigations.

²⁶ "All-Welded Tanker *Phoenix*," by W. A. Stewart, Trans. North East Coast Institution of Engineers and Shipbuilders, vol. 62, part 2, December, 1945, pp. 35-60.

²⁷ Department of Engineering, University of Cambridge, Cambridge, England.

The first is the effect of test-piece depth on breaking loads. The author uses specimens of the type shown in Fig. 31 herewith, varying d and keeping a constant. He obtains a curve, Fig. 32 of

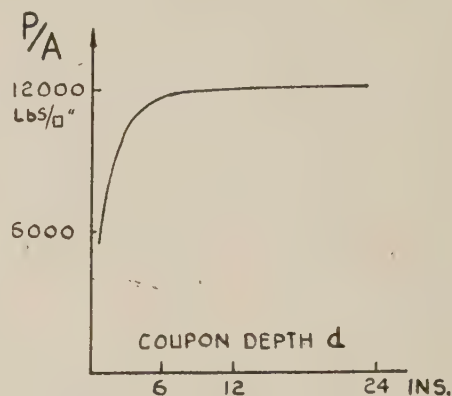


FIG. 32

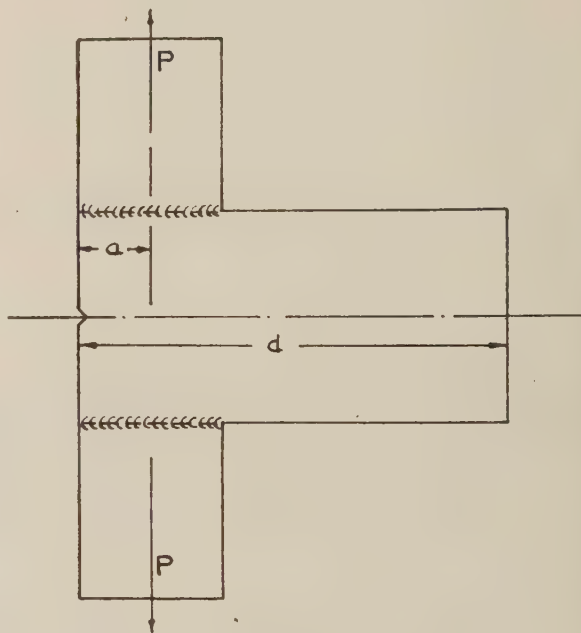


FIG. 33

this discussion, which shows that the maximum load/unit area P/A , for brittle fractures, is small for small values of d , and that it increases up to a value of d of about 6 in., and then remains constant. He also produces Equation [10] which represents this result analytically.

$$\frac{P}{A} = \frac{48,000}{1 + 0.617 \left[\frac{6e}{d-h} + k(d-h-6) \left(\frac{e}{d-h} - 0.50 \right) \right]}$$

where $n = 0.617$ is a constant introduced on account of the notch; $k = 0$ when $d-h < 6$ in., and $k = 1$ when $d-h > 6$ in., e, d, h being defined in Fig. 31; $S = 48,000$ psi is the breaking strength of a specimen with a notch but with no eccentricity.

In carrying out tests with specimens of the type shown in Fig. 33 herewith, however, it has been found that by decreasing d and keeping a constant, the P/A value increases.²⁸

This apparent anomaly can be explained as follows: The effect

²⁸ Report to Admiralty Ship Welding Sub Committee, by C. F. Tipper, FE4/181, September 1946.

of d upon the strength P/A , depends upon the value of a . This is not pointed out by the author although it is implied in his analysis. Substituting $e = a + (d + h)/2$ in his Equation [10] and putting $k = 0$, since P/A only changes when $d - h < 6$ in., and using symbols instead of numerical values

$$\frac{P}{A} = \frac{S}{1 + n \left[3 + \frac{6}{d - h} (a + h) \right]}$$

Fig. 34 of this discussion shows how the curve is reversed when $(a + h)$ is negative, which is the case when the line of action of P

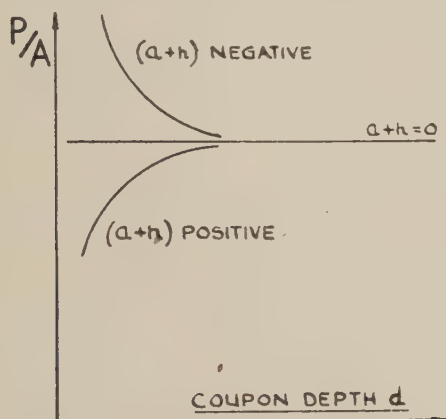


Fig. 34

is inside the section as in test pieces of the type shown in Fig. 33. When $(a + h) = 0$, P/A does not change with depth of test piece. The eccentricity of loading can be measured by either e or a . If the depth is altered, one of these must change, so that the "eccentricity" is not kept constant.

A further point which needs some clarification is the significance of the factor n .

The author points out in Equation [1] that the tensile stress in an eccentrically loaded specimen without a notch is

$$S = \frac{P}{A} + \frac{Mc}{I}$$

where c is the distance of the extreme fiber from the center line. This formula is then modified to account for the introduction of a notch, giving his Equation [3]

$$S_N = \frac{P}{A} + nq \left(\frac{Mc}{I} \right)$$

where

S_N = breaking strength of rectangular section with specified notch with no eccentricity

n = "correction factor"

q = anisotropy factor, assumed unity for fractures transverse to direction of rolling

It might seem at first that the notch only affects true bending stresses and not the axial stresses but it is important to realize that in Equation [3] stresses are no longer being considered, but have been replaced by breaking strengths of certain types of specimen under certain specified load systems. Equation [3] might have been written

$$S_N = n_1 \frac{P}{A} + n_2 \left(\frac{Mc}{I} \right)$$

where n_1 and n_2 are actual stress-concentration factors for pure tension and pure bending, respectively, but on account of the definition of S_N , n_1 can be taken as unity. There is no reason why n_2 should be equal to n_1 ; Frocht,²⁹ in photoelastic analyses of stresses in a grooved bar, obtains stress-concentration factors of 2.90 for pure tension and 2.21 for pure bending. Although it is not possible to compare breaking strengths with stresses in an ideal elastic body, it is interesting to compare the author's value of $n = 0.617$ with Frocht's ratio $n_2/n_1 = 0.76$, showing at least a qualitative agreement.

The second point at variance is that the author finds a decrease in the value of P/A as the fracture changes from fibrous to brittle, whereas in notch-tensile and notch-bend tests³⁰ no such change occurs. A slightly higher maximum load is reached in such tests as the temperature falls whether the final fracture is crystalline or fibrous.²⁸ Compared with these, a lower value is usually obtained with 3-in-wide strips, with one notch, accompanied by a slight fall when the fracture changes in character,²⁸ as is also the case with test pieces of the shape shown in Fig. 33. The loads required to break such test pieces in a ductile manner are only slightly higher when the fracture is fibrous.

It may be assumed that the sequence of events up to the beginning of the fracture is the same in both test pieces, whether a brittle or a fibrous fracture occurs. More plastic deformation takes place at the notch, the more ductile the fracture. However, this is true of both tensile and bend tests which reach the same load before fracture. Therefore the higher loads of test pieces which break with a fibrous fracture in the author's test and in 3-in-wide notched tensile tests must be attributed to a condition arising after the crack has started.

In brittle fractures the load drops immediately owing to the large number of crystals which cleave. In the fibrous type of fracture the load may continue to rise, due to plastic deformation and hardening, even though the section is being gradually reduced in area as the crack spreads. With small sections the effect is not pronounced but as the width increases, the load to fracture may increase until it reaches a value comparable with the ultimate breaking strength of the material before it falls as the cross section is reduced.

With test pieces shaped as in Fig. 33 of this discussion, which failed in a ductile manner, the fracture path did not long remain perpendicular to the direction of loading. It sheared off to the point of attachment of the grips, so that little increase in load would be expected after the fracture had started.

Although the author's results appear to disagree in some respects with those obtained in the Engineering Department, Cambridge University, the difference observed can be explained by the different forms of test piece used and hence the different method of loading.

AUTHOR'S CLOSURE

The author wishes to express his appreciation to the discussers for their interest in the paper and for their contributions.

In the treatment of a number of important items commented upon by several of the discussers, considerable additional test data are presented in this closure. In obtaining these data the testing procedure and conditions were the same as those described in the paper, under the section, "Determination of Effect of Variables on Development of Cleavage Fractures." Unless otherwise specified, the steels were tested in the "as-received" condition with the use of the D-type coupon shown in Figs. 1 and 37, containing the standard 45-deg pressed notch of $3/16$ in. depth

²⁹ "Photoelasticity," by M. M. Frocht, John Wiley & Sons, Inc., New York, N. Y., 1941, pp. 235-237.

³⁰ "Slow Bending Tests on Large Notched Bars," by J. G. Docherty, *Engineering*, vol. 139, 1935, pp. 211-213.

and 0.0015 in. root radius. For notches with root radii other than 0.0015 in. the same depth and angle were used as in the standard notch, except in the case of notches of 0.039 in. root radii where the Charpy keyhole type of notch of $\frac{3}{16}$ in. depth was used. Notches with root radii larger than 0.0015 in. were made by milling instead of pressing. In preparing charts, the average values instead of individual test results were plotted, unless otherwise stated. The fractures in all cleavage-tear as well as Charpy impact tests were transverse to the direction of rolling of the plate. All Charpy impact tests were made with the use of the standard 10-mm square coupon containing the keyhole notch of 0.039 in. root radius.

J. G. Althouse states that tests for determination of transition temperatures and any other test are of questionable value for evaluating quality of steels in the term of service performance.

It does not appear to the author to be justifiable to assume that steels possess an inherent property of fracturing in service in an unpredictable manner. It is agreed that the evaluation of service performance may be incomplete or even misleading if it were based on transition temperatures alone, especially if the test coupon used for such tests has an arbitrary geometry and is not related to the loading conditions or to the structural rigidity obtained in service. It is believed that the testing procedure described in the paper and elaborated upon in this closure, simulate some of the basic conditions encountered in service. If it were not possible to make some evaluation of the quality of steels by tests, Mr. Althouse's statement regarding superiority of certain steels in service performance over other steels could not be substantiated.

W. Bennett discusses from a practical point of view the failure of ships through development of cleavage or brittle fractures, inspection of ships during construction, and the quality and testing of steels. It is to be realized that fracturing or breaking of ships of all-welded construction cannot be attributed to a single cause or condition. Such failures appear to occur only when several conditions co-exist, or prevail simultaneously, in the critical sections of a ship. These conditions include the following:

(a) The sum of static, dynamic and other stresses imposed or developed must be sufficiently high to initiate fracturing. Surges or impulses of dynamic, impact, or thermal stresses are not necessary to start a cleavage fracture. However, if present they will induce fairly high stresses, which add to the static stresses.

(b) Discontinuities, defects, or notches must be present, of which at least one conforms to certain geometrical shapes and is located in a definite orientation with respect to the principal stress. For example, there would be less likelihood of developing large cracks from a small crack or a sharp weld undercut running "parallel" to the direction of the principal stress, than from a similar discontinuity running "transversely" to the principal stress. Furthermore, if these discontinuities or notches have the proper orientation but are located more than a certain distance away from the area which carries the peak of the stress, it would be improbable that a fracture would initiate from them.

(c) The steel used must be incapable of yielding or deforming to the extent necessary for reducing the principal stress, under the state of stress and at the temperatures encountered in service. That is, the temperature of the steel is considerably below its actual transition temperature range, corresponding to the rigidity and massiveness of the structure and to the type of notch present.

On this basis, no fracturing of welded ships would be likely in the absence of any one of these conditions. Coincidence of all these conditions is undoubtedly the exception rather than the rule. It is probably for this reason that only a relatively small percentage of all-welded ships have failed by cleavage fracturing.

Condition (a) is related to service peculiarities which can be

controlled only to a small extent. However, it is also related to the design details of the ship, since the stresses in the critical sections can be reduced materially by the use of higher factors of safety, which is discussed in the paper, and by other design modifications treated in the section of this closure dealing with E. M. McCutcheon's discussion.

If discontinuities are present, including those created by design or workmanship, condition (b) cannot be controlled. Complete elimination of defects or discontinuities is not likely to be accomplished in a complicated structure like a ship by x-ray inspection, due to certain inherent limitations of this method of inspection, although psychologically and actually the x-ray inspection could be of considerable help for improving the quality of welding. For example, structural fillet welds cannot be x-rayed satisfactorily, of which type of welds there is a large number in the critical sections of a ship, and because all defects in the butt welds cannot be detected by x rays. Magnaflux and similar inspection methods are very useful for detection of surface cracks, small notches, and related defects. Careful visual inspection by experienced inspectors or surveyors, and proper technique and sequence of welding are of course very important, as discussed by Bennett.

The service temperature of a ship cannot be controlled, but the quality of steel, listed as condition (c), could be controlled. However, no definite or economical path has as yet been indicated in the direction of modifying the quality of hull steel for construction of all-welded ships. The absence of a generally accepted testing method and of a concept as to properties sought has complicated the simplification of the problem. If conditions (a) and (b) prevail, considerable improvements in the quality of steel seem to be necessary, in order to immunize completely a welded ship against development of large cleavage fractures, under all conditions of service, stress, temperature, and stress concentrations caused by structural or inherent discontinuities. In this connection, attention is called to the fact brought out by this investigation, that in the presence of sharp notches the actual load-carrying capacity of a steel decreases materially if the temperature is below its transition range established by the geometry of the notch. Apparently this fact had not been generally recognized heretofore. Substantial improvements in the quality of steels appear also necessary in this light, in the direction of increasing their load-carrying capacities at low temperatures and in the presence of sharp notches.

The British experience in the matter of the manganese/carbon ratio is helpful. Nevertheless, tests made by the author on a semikilled plate of 1 in. thickness and with a manganese/carbon ratio of slightly over 3 indicated no noticeable superiority of this steel in the "as-rolled" condition over other semikilled plates of the same thickness and with a manganese/carbon ratio not much over 2. This is steel 18, the chemical and physical properties of which are listed in Table 14. Its transition temperatures, determined by the Charpy impact and cleavage-tear tests, are given in Tables 15 and 16, respectively. Steels with a manganese/carbon ratio not less than 3 have been used for construction of pressure vessels. Difficulties of welding rigid sections appear to increase as the manganese/carbon ratio reaches about 4, especially if the carbon content is more than about 0.20 per cent and if the welding is done without preheating. Some increase in the manganese/carbon ratio may be a move in the right direction, such as adopted by the 1948 Rules of the American Bureau of Shipping. However, it appears questionable whether the mere increase of 0.10-0.20 per cent of manganese or the proportionate percentage of decrease in the carbon content is the final answer. In order to find an economical solution, several other aspects of the problem should be considered. The latter should include a comparison of the cost involved in the use of improved quality of steel to the cost of increasing the factor of safety of the critical

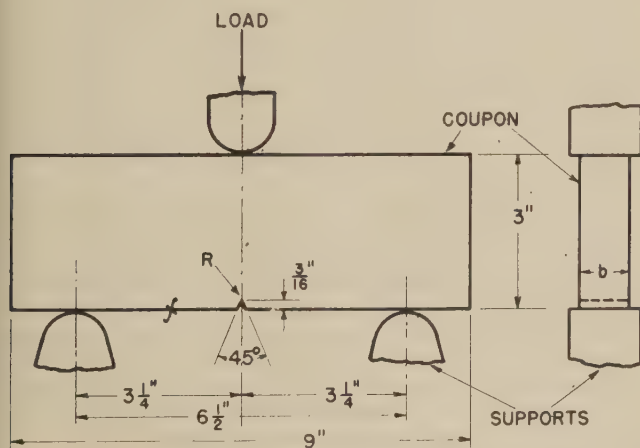
section of a ship through modifications of design, and details related thereto.

TABLE 14 CHEMICAL AND PHYSICAL PROPERTIES OF STEELS 18, 19, AND 23

Plate thickness, in.	1	Steel 18	1	Steel 19	15/16	Steel 23
Chemical composition, per cent:						
Carbon.....	0.20		0.17		0.25	
Manganese.....	0.63		0.44		0.45	
Phosphorous.....	0.020		0.014		0.015	
Sulphur.....	0.033		0.023		0.024	
Silicon.....	0.07		0.17		0.05	
Tensile properties (fractures transverse to rolling):						
Tensile strength, psi..	61300		65700		63500	
Yield, point, psi.	31400		36500		33500	
Elongation in 8 in. per cent.....	34.4		32.3		31.7	
Grain size, normality, and inclusion contents:						
Actual grain size.....	5-7		7-8		5-6	
Austenitic grain size.....	3-4		6		6-7	
McQuaid-Ehn normality.....	Normal		Normal		Normal	
Inclusion content and type.....	A heavy 1 and D thin 3		A heavy 1 and D heavy 2		C thin 2 and D heavy 2	

NOTE: The chemical and physical properties of steels 20 and 22 are given in Tables 2 and 3 of the paper.

There is no standard or generally accepted testing procedure for evaluating the resistance of steels to cleavage fracturing. Just what testing procedure is to be used depends on what properties one desires to determine or evaluate and with what degree of accuracy. As described in the paper, in the initial stages of this investigation the test coupon used was a simple beam, notched at its edge and broken by a concentrated load at the center, with the beam being supported at the two ends. This test coupon is shown in Fig. 35 herewith, and is designated DX-type. As it became



"b" = full plate thickness.

"R" = notch root radius = 0.0015", or as specified.

FIG. 35 SIMPLE BEAM OR DX-TYPE TEST COUPON

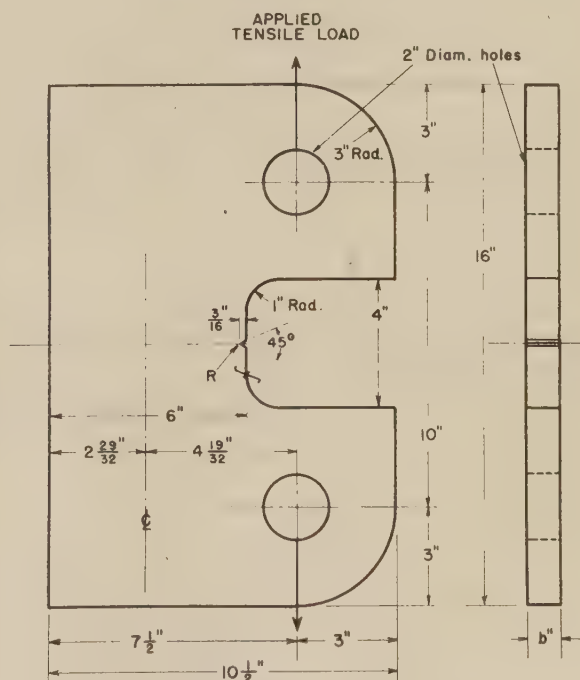
necessary to investigate the effects of section and eccentricity variations, it was found necessary to modify this coupon. The first modification introduced was to weld two arms to the notched edge of the beam, by means of which arms the specimen could be pulled in a tensile testing machine, instead of inducing tension in the notched edge by a pushing force as in the beam coupons. This modification produced the test coupon of the type shown in Fig. 36, and permitted testing of sections of somewhat larger sizes. The particular test piece shown in this figure is approximately 1 in. thick $\times$ 11 in. wide $\times$ 12 1/2 in. deep, in which a cleavage fracture initiated from the notch and penetrated to a



FIG. 36 TEST COUPON INTERMEDIATE BETWEEN TYPES DX AND D

depth of about 2 in., changing to shear as the fracture progressed. However, the loading arms, being of the fixed type, were found to introduce auxiliary stresses and eccentricities of unknown magnitudes, and thereby to interfere with the determination of breaking and yield loads and other characteristics with sufficient accuracy. For this reason, the use of test coupons of this type was abandoned by the author about 4 years ago. It came to the author's attention only recently that the test coupon used by Dr. Tipper and associates, Fig. 33, is similar to the type just discussed.

The type of test coupon finally adopted by the author is shown in Fig. 1, and the dimensional details and the notch geometry of the D-type coupon used as the standard are given in Fig. 37 of this closure. It is to be noted that it has a "hinged" instead of fixed-arm loading, by virtue of which breaking and yield loads



"b" = full plate thickness.

"R" = notch root radius = 0.0015", or as specified.

TYPE "D" COUPON

FIG. 37 STANDARD CLEAVAGE-TEAR TEST COUPON, DESIGNATED D-TYPE

can be determined more accurately. This coupon was found to be suitable for determining the load-carrying capacity, transition temperatures, geometry effects, and behavior of steels subjected to multiaxial stresses. It was not chosen arbitrarily but established by a large number of tests given in the paper.

If arbitrary transition temperatures are to be determined without evaluating the effects of size and notch-geometry variations and the load-carrying capacity, the conventional impact tests could be used. A correlation of the transition temperatures, determined by the latter tests with those of cleavage-tear tests, is given in this closure in connection with C. L. Clark's discussion. The DX-type coupon can be used for routine work. This test coupon does not have the usefulness and accuracy of the D-type coupon, but is suitable for determining approximate breaking loads as well as transition temperatures. It is considerably cheaper to prepare than the Charpy impact and the D-type coupons, and can be made from the duplicate blanks provided at steel mills for making bend or "nick-break" tests on steel plates.

There are many other proposed test coupons of various geometries, and numerous testing methods, the discussion of which is beyond the scope of this closure.

The work done in the United Kingdom on these problems is well-recognized. The particular references mentioned by Bennett were published considerably later than the submittal of the author's paper to this Society for presentation, for which reason they could not be cited in the paper as references. Aside from this, the literature on fractures is so voluminous that only those references are cited by the author from which direct quotations have been made.

C. L. Clark inquires about tests in the fusion and gradient zones of welds. In this investigation the notches were located in the weld-deposited metal, since the object was to determine the type of fracture obtainable in the weld-deposited metal. Unquestionably, exploration of the fusion area and the heat-affected zone in the parent plate should yield interesting results.

In his discussion, Mr. Clark comments on the significance of the fact that the steels tested were classified in the same order by the cleavage-tear and the Charpy impact tests. In addition to obtaining a similar classification of steels with respect to their transition temperatures, a correlation can be established between the transition temperatures determined by these two test methods³¹ by the use of the following equation

$$T_c = T_0 \left[1 - C \left(\frac{1}{R} \right)^d \right] \dots \dots \dots [22]$$

³¹ "Cleavage Fracturing and Transition Temperatures of Mild Steels," by A. B. Bagsar, *The Welding Journal*, vol. 27, 1948, Research Supplement, pp. 123-s-131-s.

where

T_c = transition temperature, in deg F abs, determined by Charpy impact test, using keyhole standard notch

T_0 = transition temperature, in deg F abs, determined by D-type cleavage-tear test coupon with a notch root radius R in.

C and d = constants for a given steel and test conditions

Since T_c is a constant for a steel in a given condition, it is apparent that Equation [22] is of the type: $x = Ky^n + A$, in which x and y are the variables. The validity of Equation [22] can then be established if it can be shown that T_0 is a function of R , of the type indicated by Equation [22]. This is proved to be the case by the data included in this closure. Accordingly, it is also possible to establish a relationship between the transition temperatures determined by the cleavage-tear tests and those determined by the use of coupon, section, and notch geometries other than those of the standard Charpy coupon. Thus the transition temperatures determined by the Izod coupon or by coupons containing the Izod notch can be correlated to the cleavage-tear test transition temperatures. To obtain this correlation it is necessary to determine the T_0 versus R curve of the steel in question by cleavage-tear tests, as shown in Fig. 41 herewith, for steels 18, 22, and 23, by means of which the numerical values of the constants in the right-hand term of Equation [22] can be found.

Assuming $T_c = T_0 C \left(\frac{1}{R} \right)^d$, Equation [22] can be written in the following form

$$T_c = T_0 - T_s \dots \dots \dots [23]$$

According to Equations [22] and [23] the numerical values of T_0 and T_s change as the notch root radius is changed, since T_s is a constant for a steel in a given condition; also for a given R the value of T_s should be approximately the same for steels of the same general class unless the slopes of the R versus T_0 curves of the steels are radically different from each other. It follows then that T_s as well as T_0 are useful indexes for evaluation of steel quality. The data discussed in the paragraphs immediately following confirm this.

The transition-temperature determinations by the Charpy impact and cleavage-tear tests of the four steels initially tested are presented in Tables 11 and 12, and Fig. 15 of the paper. Similar data are given in Tables 15 and 16, and Fig. 38, of this closure, on five additional steel plates of different heats and thicknesses. These include steels 20 and 22 of 2 1/4 in. and 1/2 in. thickness, respectively, the chemical and physical properties of which are given in Tables 2 and 3 of the paper; also steels 18, 19, and 23, of approximately 1 in. thickness, the chemical and physical proper-

TABLE 15 DETERMINATION OF TRANSITION TEMPERATURES BY CHARPY IMPACT TESTS; KEYHOLE-NOTCHED STANDARD COUPONS

Test temp deg F	Impact values in ft-lb									
	Steel 18	Average	Steel 19	Average	Steel 20	Average	Steel 22	Average	Steel 23	Average
-80	...	2	...	...	...	...	...	...	...	...
-50	...	6, 2, 15	...	...	...	...	4, 2	...	...	...
-40	...	11, 2, 20	...	...	...	...	...	...	...	...
-30	...	21, 22, 11	...	...	...	...	...	...	...	...
-20	3, 6	4	...	...	2, 2	2	5, 6	5	...	...
-10	10, 10	10	...	...	6, 3	4	11, 14, 19	12	2, 3	2
0	18, 24, 17	20	24, 24	24	5, 8, 18	10	19, 15	17	5, 5	5
15	23, 17, 27	22	...	...	27, 28	27	...	...	21, 10	15
32	23, 18	21	25, 26	25	29, 31	30	21, 21	21	23, 18	20
45	31, 34	32	...	...	...	...	...	...	26, 24	...
60	...	...	...	...	35, 33	34	...	...	29, 30	30
85	32, 34	33	32, 32	32	38, 39	39	22, 24	23	32, 30	31
140	...	...	36, 37	37	38, 40	39	...	...	...	...
Transition Temps—deg F	5		-38		10		-15		20	

NOTE: In these tests as well as in those listed in Table 12 of the paper, all Charpy coupons were cut from the mid-thickness section of each plate and the notches drilled at right angles to the rolled surface of the plate. In all cases, fractures were transverse to the rolling direction of the plate. Transition temperature = temperature corresponding to point representing the average of the maximum and minimum impact values on the energy versus temperature curve.

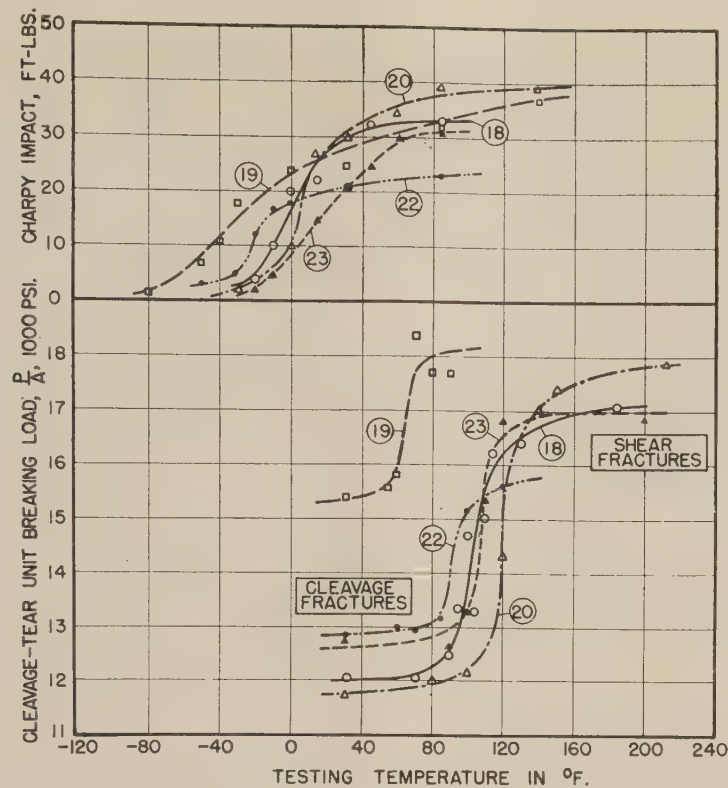


FIG. 38 TRANSITION CURVES OF $\frac{1}{2}$ IN.- TO $2\frac{1}{4}$ -IN-THICK STEEL PLATES, DETERMINED BY STANDARD CHARPY AND CLEAVAGE-TEAR TESTS

ties of which are given in Table 14. In Tables 15 and 16 are summarized test data for determination of transition temperatures of these steels by the Charpy impact and cleavage-tear tests, respectively. The transition curves are shown in Fig. 38. Included in Table 16 are test data on steel 18 for the D-type coupons as well as for coupons of 10 in. depth. Since the unit breaking loads and the transition curve obtained for the 10-in.-deep coupons are practically the same as for D-type coupons, the P/A values for the 10-in. coupons are not plotted in Fig. 38.

In Table 17 are summarized the transition temperatures, as well as the T_s values for the nine steels mentioned. It is to be noted that with the use of the standard notch the average difference between the Charpy and cleavage-tear test transition temperatures is approximately 100 deg F for all the steels tested. This difference of 100 deg F, or T_s , as defined by Equation [23],

TABLE 17 COMPARISON OF TRANSITION TEMPERATURES OF NINE OPEN-HEARTH STEEL PLATES DETERMINED BY CHARPY IMPACT AND CLEAVAGE-TEAR TESTS

Steel	Transition temperatures deg F			
	Thickness, in.	Charpy impact	Cleavage- tear	T_s in deg F (see note)
Rimmed steels:				
Steel 7.....	1	17	107	90
Steel 22.....	1/2	—15	92	107
Semikilled steels:				
Steel 15.....	1	18	103	85
Steel 17.....	1	—17	83	100
Steel 18.....	1	5	105	100
Steel 23.....	15/16	20	110	90
Killed steels:				
Steel 16.....	1	—32	68	100
Steel 19.....	1	—38	65	103
Steel 20.....	2 1/4	10	120	110
Average				100

NOTE: T_s = Difference, in degrees F, between the transition temperatures determined by the Charpy impact (keyhole-notched coupon) and cleavage-tear (D-type coupon with 0.0015-in. notch root radius) tests

TABLE 16 DETERMINATION OF TRANSITION TEMPERATURES BY CLEAVAGE-TEAR TESTS (D-type coupon, 45-deg notch of $\frac{3}{16}$ in. depth and 0.0015 in. root radius)

Test temp, deg F	Steel 18, 1 in. (see note)		Steel 19, 1 in.	Steel 20, 2 ¹ / ₄ in.	Steel 22, 1 ¹ / ₂ in.	Steel 23, 1 ⁵ / ₁₆ in.
—40	12000	...	...	...	...	11600
—10	...	...	...	...	...	12400
20	...	...	...	...	13300	...
32	12000-12100	12400-14000	15000-15800	11500-12000	12700-13000	12300-13300
55	...	...	15600	...	...	...
60	...	13100	14000-17500	...	13000	12100
70	11800-12300	...	18300-18500	...	12800-13100	...
80	12600	...	17700	12000	...	12600
90	11800-13300	13600	17700	...	13200	12600-12700
95	13500	12200-13000	...	...	...	...
100	12300-16500	...	...	12100-12300	15000-15300	12700-14200
105	12500-14300	16400	...	...	...	...
110	15000	17400-17600	...	...	...	13500-17200
115	15500-16700	17300-17400	...	...	...	...
120	...	16000	...	13200-16000	15600	16000-17600
130	16400	...	...	...	...	...
140	...	...	...	17000	...	16000-17600
150	...	...	...	17400	...	...
186	17100	17200	...	...	15300	...
200	...	...	...	...	...	16700
212	...	...	...	17900	...	...
Transition temp., deg F	105	...	65	120	92	110

NOTE: Type-12 coupon (see Fig. 1 of paper), except with a section depth of 10-in. instead of 12-in., containing the notch described.

is practically independent of the thickness of the plate or of coupon depths beyond 6 in., within the thickness range of $1\frac{1}{2}$ to $2\frac{1}{4}$ in. investigated. At first glance it would appear that the value of T_s , corresponding to the standard notch used, is likewise independent of the quality of the steels tested in the as-rolled condition, since the nine steels investigated include two rimmed, four semikilled, and three killed steels. However, it will be shown in connection with the discussion of Fig. 41 that the value of T_s is not the same for the steels if notches other than the standard notch were used. It is also apparent that the value of T_s changes if the steel is heat-treated, an alloy type, or contains an abnormal amount of impurities.

Equation [23] of this closure indicates that the temperature at which a given steel changes from shear to cleavage type of fracturing is determined by two main factors. The first factor is inherent in the steel and is its homogeneity or character, which can be expressed by an arbitrary or relative index obtained, say, by the conventional impact tests, such as the Charpy impact test. The other factor is likewise inherent in the steel and represents the response of the steel to the multiaxial stresses created by the notch. As discussed in the paper, the state of stress is determined by the geometry of the notch, the stress concentration increasing as the notch root radius is decreased, with other test conditions being the same. Therefore the cleavage-tear test expresses the combined effects of both of these two factors in terms of the yield and fracture loads.

H. G. Enzian comments on the advantage of the use of the standard Izod notch in cleavage-tear tests. It appears to the author that the choice of the notch to be used should be based upon experimental or rational data, rather than upon arbitrary standards, as the data presented herewith show. The quantitative effect of variations of notch root radii on the transition temperatures and breaking and yield loads of steels 18, 22, and 23 are given in Tables 16 and 18, and graphically presented in Figs. 39

to 42. In these tests, in order to obtain greater accuracy, most of the yield and breaking loads were determined by the photographic method, the details of which method are described in the discussion of Fig. 50 of this closure. In this connection it should be mentioned that where notches of large root radii are used, the transition temperatures cannot be located definitely by the appearance of the changes in the fracturing characteristics. Such changes in the fracture appearance may occur at temperatures considerably above the transition temperatures indicated by the change in the yield and breaking loads, as is shown in Figs. 39 and 40, to be the case for steels 22 and 18, respectively.³²

Figs. 39 and 40 of this closure show the effect of variations of R on the transition temperatures of steel 22, and of steels 18 and 23, respectively, which were obtained by plotting the data given in Tables 16 and 18. It is to be noted that the unit breaking loads P/A , for cleavage are lower than those for shear, if the notch root radii used are 0.0075 in. or smaller. It appears that in notches with root radii greater than about 0.0075 in. the extent of stress concentration is not sufficiently high to induce cleavage fracturing; fractures initiating from such notches occur by shearing or yielding. Notches of 0.0075 in. or smaller root radii appear to be capable of concentrating stresses to a sufficient degree so that the notch root radii of the latter dimensions become the main determinants of the fracturing loads. The Izod notch ($R = 0.01$ in.) occupies an intermediate position, and in the author's opinion, therefore, should not be chosen as the standard.

It is apparent from these data that the transition ranges of steels can be shifted to higher temperature levels by decreasing the notch root radii and vice versa, other conditions being the

³² In a recent paper by the author³¹ the transition temperatures of steels 18 and 22 for the Charpy notch ($R = 0.039$ in.) were estimated by the change in fracture appearance. The procedure described in this closure is more accurate.

TABLE 18 EFFECT OF NOTCH ROOT RADIUS R , ON UNIT BREAKING AND UNIT YIELDING LOADS AND ON TRANSITION TEMPERATURES, T_0

(a) Unit Breaking Loads, P/A , psi									
Test temp. deg F	$R = 0.0008$ in.		$R = 0.0075$ in.		$R = 0.017$ in.		$R = 0.039$ in.		
	Steel 22	Steel 18	Steel 22	Steel 23	Steel 22	Steel 23	Steel 22	Steel 18	
-40	...	...	...	...	17000	...	19300	...	
-10	...	...	...	...	16900-17500	...	18500	...	
20	...	...	...	...	17300	...	18500	19000	
32	12300-13000	11300-11600	15400-15500	13700-14000	16600	15400	18000-18800	18000-18800	
40	...	...	...	...	15700-16700	...	17700-18600	18800	
50	...	...	15300-15700	...	16700-17000	...	18000-18000	...	
55	...	...	15000-15500	...	...	...	...	...	
60	...	...	17000-17300	14200	...	15600-15800	...	19000	
70	12700	12000	16700-16800	14000	16200-17000	15400-16300	17500-17700	17500-17700	
80	13000	11500	...	13300-14700	...	14800-14800	17400-17700	17400-17700	
90	...	...	17000	13100-14000	...	14500	...	...	
95	...	...	...	14200	...	...	17600	17600	
100	13500	11700-12100	...	14800	...	14400	...	...	
110	15000-15200	...	...	14700	...	...	...	...	
120	14500-15000	12000-15800	...	...	...	...	...	...	
130	15000-15100	13300-15600	...	...	...	...	...	...	
140	...	15000-16300	...	...	...	...	...	...	
150	...	15200-15800	...	14700	...	14500	...	...	
186	15100	...	...	...	...	...	...	...	
212	...	...	...	...	16000	...	17000	17000	
	$T_0 = 105$ deg F	$T_0 = 125$ deg F	$T_0 = 58$ deg F	$T_0 = 90$ deg F	$T_0 = 45$ deg F	$T_0 = 80$ deg F	$T_0 = 30$ deg F	$T_0 = 50$ deg F	

(b) Unit Yield Loads, Y/A , psi										
Test temp. deg F	$R = 0.0015$ in.			$R = 0.0075$ in.		$R = 0.017$ in.		$R = 0.039$ in.		
	Steel 22	Steel 18	Steel 23	Steel 22	Steel 23	Steel 22	Steel 23	Steel 22	Steel 18	
-40	...	12000	11600	...	...	14700	...	15300	...	
-10	...	...	10700	...	...	15300	...	15200-15500	...	
20	13300	...	...	...	...	14000	...	14300-14300	...	
32	...	12150	9000-9400	...	14500	14000-14300	...	13200-14600	...	
40	...	...	...	...	13700	13300-14000	...	13200-13300	...	
50	...	...	...	...	14500	13300-14000	...	13300-13700	...	
55	...	...	...	...	13700	...	...	...	...	
60	...	...	7600	...	13800	...	13600	...	...	
70	...	...	...	...	...	...	...	13000-13300	...	
80	...	9300	7800	...	...	...	...	13300	...	
90	...	...	...	13700	...	...	...	...	...	
95	...	10000	...	...	...	...	...	...	...	
100	13300	...	8000	...	...	...	...	13000	...	
110	13200	9800	7000	...	...	...	...	...	...	
120	13300	9500	...	...	...	...	...	...	...	
186	12700	...	...	...	...	...	...	...	...	
212	...	...	8550	...	...	10300	...	11000	...	

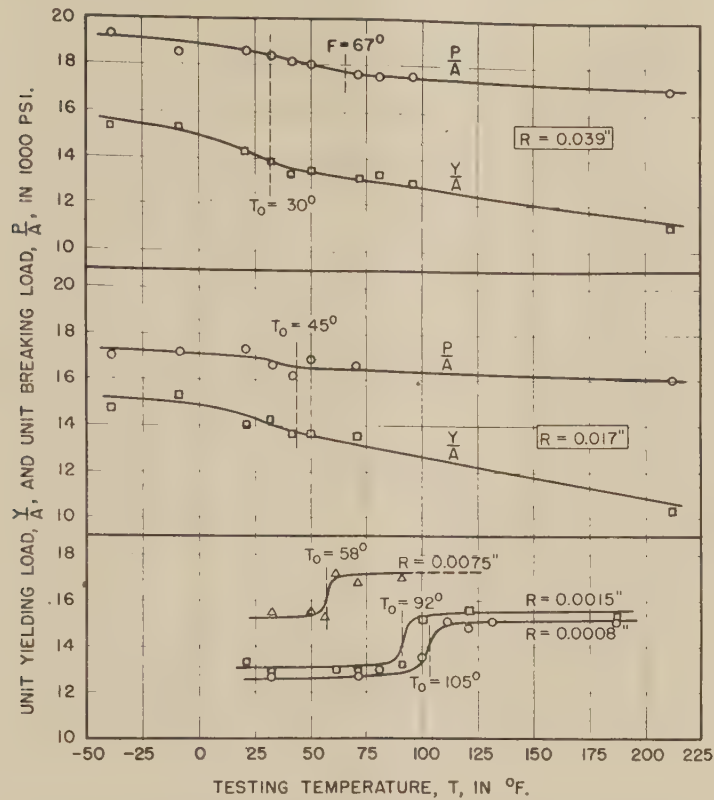


FIG. 39 TRANSITION CURVES OF STEEL 22 FOR NOTCH ROOT RADII INDICATED

(Transition temperature = T_0 , change in fracture appearance = F . The three curves in bottom section are P/A versus T curves for the steel.)

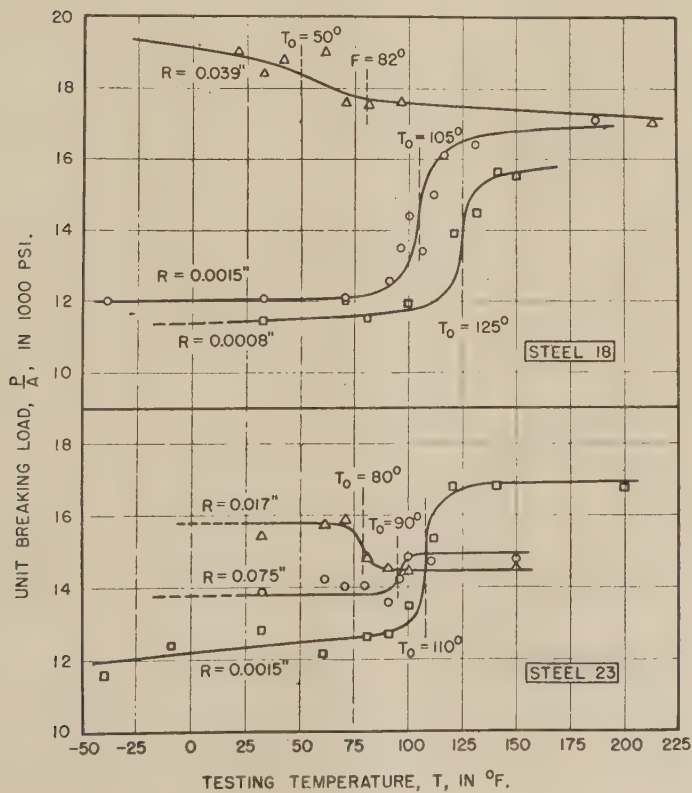


FIG. 40 TRANSITION CURVES OF STEELS 18 AND 23 FOR NOTCH ROOT RADII INDICATED

(Transition temperature = T_0 , change in fracture appearance = F .)

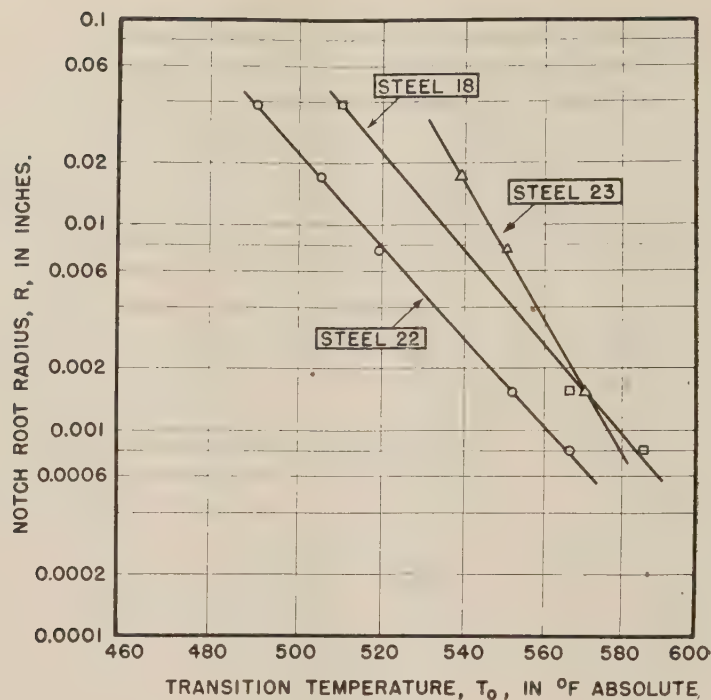


FIG. 41 TRANSITION TEMPERATURES OF STEELS 18, 22, AND 23 AS FUNCTION OF NOTCH ROOT RADIUS

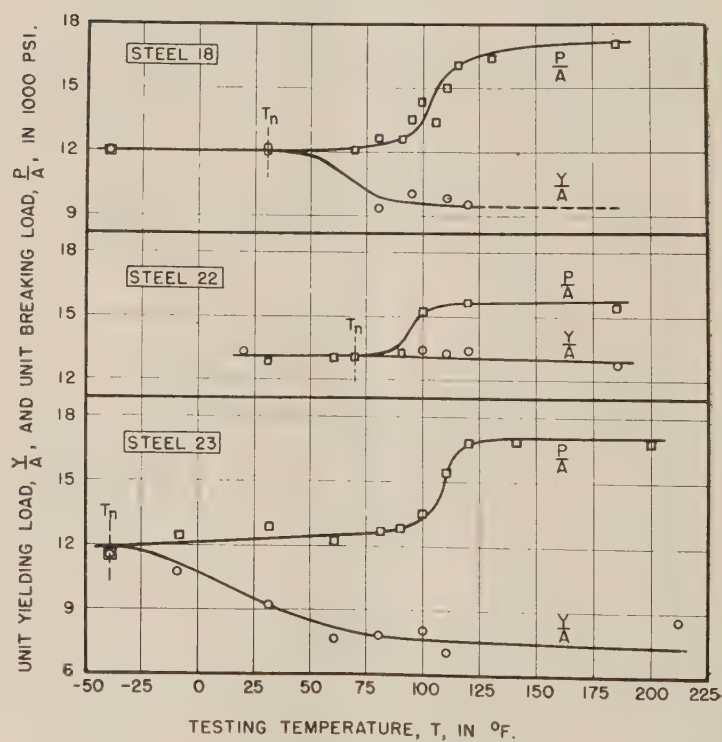


FIG. 42 UNIT BREAKING LOADS P/A AND UNIT YIELDING LOADS Y/A , OF STEELS 18, 22, AND 23 AS A FUNCTION OF TESTING TEMPERATURE

(Test coupon used D-type, containing standard notch. At temperature T_n the P/A and Y/A coincide with each other).

TABLE 19 TRANSITION TEMPERATURES OF STEELS 18, 22, AND 23 FOR VARIOUS NOTCH ROOT RADII

Notch root Radius R , in.	Steel 18		Steel 22		Steel 23	
	Transition temp, deg F	T_s , deg F	Transition temp, deg F	T_s , deg F	Transition temp, deg F	T_s , deg F
3×10^{-8}	200 ^a	195 ^a	190 ^a	205 ^a	170 ^a	150 ^a
0.0008.....	125	120	105	120	...	...
0.0015.....	105	100	92	107	110	90
0.0075.....	...	...	58	73	90	70
0.017.....	...	...	45	60	80	60
0.039.....	45	40	35	50	...	...
Charpy transition temperatures, deg F	5	...	-15	...	20	...

^a Extrapolated values

same. In Table 19 are summarized the transition temperatures of steels 18, 22, and 23, corresponding to root radii varying from 0.0008 in. to 0.039 in. Included in this table are also the values of T_s for these steels for various notch root radii. From the data of Table 19, Fig. 41 was constructed, which is a logarithmic plot showing the transition temperatures, in deg F abs, as a function of the notch root radii. As required by Equations [22] and [23], the resultant curves are straight lines, their equations being of the type: $T_0 = QR^a$, in which T_0 is the transition temperature in deg F abs, R is the notch root radius, and Q and a are constants for a given steel in a given test condition. As stated in the paper, the radius at the base of cleavage cracks is of the order of 3 millionths of an inch. By extending these lines to this value of R , the transition temperatures of the steels containing this type of notch were estimated. These are also included in Table 19. It is to be noted that these transition temperatures are within the range of about 170 to 205 F, and that the corresponding values of T_s are within the range of 150 to 195 F. Although these are extrapolated values, nevertheless, they do indicate that our present structural steels have transition temperatures considerably above atmospheric temperatures if they contain small cleavage cracks or similar discontinuities.

The significance of the variations of the yield and breaking loads with change of temperature is indicated in Fig. 42 which was constructed from the data given in Tables 16 and 18. In this figure the unit breaking loads P/A and unit yield loads Y/A , of steels 18, 22, and 23, obtained by the use of the standard D-type coupon, are plotted as a function of the test temperature. It is to be observed that the load required to cause yielding increases as the temperature decreases, whereas the breaking loads decrease with decrease of temperature; also that the slope of the Y/A versus T curve is different for different steels. At the temperature T_n , which is considerably below the transition temperatures of the steels, the Y/A and P/A values coincide with each other. At this temperature the entire section of the steel, considered as a unit and as represented by the test coupon, shows no permanent deformation; it ruptures by cleavage fracturing while the steels are essentially within the elastic range. Between this and the transition temperature, some yielding occurs, although the appearance of the fracture may remain essentially "crystalline." This yielding, admittedly slight, however, may be sufficient to reduce the stress peaks. The behavior of steels within the temperature range of T_n to T_0 is therefore important and appears to explain satisfactorily why more service failures occur at low temperatures.

The difference between the temperatures T_n and T_0 , as well as the extent of yielding obtainable, are dependent upon the quality of the steel and section and notch geometries used. For the standard test conditions used, the T_n temperature of steel 22 occurs at about 70 F, which is 22 deg F lower than the T_0 and 85 deg F higher than the Charpy transition temperature of this steel. The T_n of steel 18 occurs at about 32 F, which is 73 deg F below T_0 or 27 deg F above the Charpy transition temperature of this steel. In steel 23, T_n is located at about -45 F, which is 155 deg F below T_0 or 65 deg F below its Charpy transition temperature. Accordingly, steel 23 could be considered as being a

steel of better quality than steel 18. This deduction could not be reached by the transition temperatures exhibited by these two steels; in fact, if transition temperatures were used as the index of quality, steel 18 would appear considerably superior to steel 23. The temperature T_n and the extent of yielding below the transition temperature are, therefore, significant indexes of steel quality, which are determinable by the cleavage-tear test.

An examination of Figs. 39, 40, and 42 reveals that different types of fracture curves are obtained by the use of notch root radii of certain ranges. These fracture curves are shown diagrammatically in Fig. 43. Type I appears to be the most desirable fracture curve for revealing the behaviorism or characteristics of steels with change of temperature; also for revealing the location of T_n , the significance of which has been described. This fracture curve is obtained by the use of D-type coupon containing notches with root radii 0.0075 in. or smaller. Type II shows the probable shape of the fracture curve for notches of the order of the Izod notch. The fracture curve in this case is approximately a straight line. Type III is obtained with the use of notch root radii of 0.017 to 0.039 in. In this case both the P/A and the Y/A values rise at the transition temperature, the rise of the P/A values being in sympathy with the rise of the Y/A values; the transition temperature can actually be determined by the shape of the Y/A versus T curve. Type IV corresponds to the fracture curve obtained with the use of the standard 0.505-in-diam unnotched tensile coupon,³³ which was included for comparison. In the latter case T_0 is depressed to temperatures below about -300 F.

It is obvious from the data just presented and discussed that in order to evaluate the behaviorism of a given steel it does not suffice to determine its transition temperature alone, especially with the use of test coupons and notches of arbitrary geometries. The determination of transition temperatures as well as the load-carrying capacity and other related important details are necessary. The ability of evaluating these characteristics is one of the main advantages of the cleavage-tear test, in addition to the other advantages discussed by Enzian. As the data show, the discriminative ability of the test is increased with the use of notches of small root radii. It is for this reason that the notch of 0.0015 in. root radius was adopted for use in cleavage-tear tests. Another advantage of this notch consists in the fact that it shows transition temperatures which are about 100 deg F higher than those indicated by the standard Charpy impact tests. Actual failures of a few ships at comparatively high temperatures indicate that the Charpy transition temperatures are lower by a margin of this order from those prevailing in structures. Additional discussion of this is given in this closure, in the treatment of A. B. Kinzel's comments.

Failures in service seldom, if ever, occur within the transition-temperature range, but usually at temperatures somewhat below this range, because some yielding is possible below this range in most steels, as has been discussed. However, the extent of this yielding is small and may be insufficient for eliminating stress

³³ "A Study of the Transition From Shear to Cleavage Fracture in Mild Steel," by H. E. Davis, E. R. Parker, and A. Boodberg, *Proceedings*, ASTM, vol. 47, 1947 pp. 483-501.

peaks effectively in some structures, although it may be sufficient to prevent failures in other structures. Therefore this margin of 100 deg F is not too conservative unless the T_n of the steel used and the state of stress involved in the intended service indicate otherwise.

The author agrees with Enzian that ship failures could be statistically analyzed for determining a definite trend, and such analyses have and are being made. However, ship failures cannot be entirely attributed to a single cause, a subject which has been discussed in connection with Bennett's comments. Fracturing of ships appears to occur under circumstances in which a combination of a number of conditions exists, a fact which necessitates careful scrutiny of failure data. As suggested by Enzian, cleavage-tear testing of steels from fractured and unfractured ships should yield interesting data. Nevertheless, ships which have failed may not necessarily be constructed of steel of inferior quality, and conversely, those that have not failed may not be made of steels of superior quality. Consequently, the extent of improvement in the quality of steel cannot be said to be in direct proportion to the percentage of ship failures. If all the conditions were favorable to fracturing, the steel of which the ship is made must be of a substantially better quality than the grades now used, in order to resist fracturing.

Finn Jonassen comments on several items dealing with transition temperatures and the breaking loads of steels. The degree of scatter within the transition range, besides being dependent upon the suitability, accuracy, and reproducibility of the test conditions and on the individuality of the steel, is determined by the law of probability. For the test coupon and notch used, the P/A versus T diagram shown in Fig. 49 represents the transition of steel from cleavage to shear, as effected by the change of temperature. It is apparent from the shape of this diagram that either mode of fracture is possible within the transition range, which range is characterized by the abrupt change in P/A and Y/A values. The curve within these temperature limits represents the average and probable values. Insufficient stock was available in the original four steels to make extensive surveys of the scatter within the respective transition-temperature ranges, since the determination of the transition temperature was only one of the variables that was to be studied in the investigation. Nevertheless, on the basis of the work done on other steels not related to the original investigation included in this paper, it is believed that sufficient data are given in Table 11 to indicate trend clearly.

The test data obtained on steel 18, which are summarized in Table 16, illustrate the degree of variations in the P/A values within the transition range of this steel, with the use of the D-type coupon containing the standard notch. Fig. 44 is the resultant transition curve, showing all the individual test results obtained. Transition curves similar to those shown in Fig. 15 of the paper, are included in Fig. 38 of this closure. The latter curves are for five different steels, representing the thickness range of $1/2$ to $2 1/4$ in., the details of which have already been discussed. It is to be noted that all these curves, including those given in Fig. 15, conform to the P/A versus T curve shown in type-I diagram of Fig. 43, and in Fig. 49.

Jonassen's second comment deals with the nominal unit breaking loads. The breaking loads reported in the paper for various tests are the values of unit breaking loads P/A , and not those of the nominal breaking strength S_N . The numerical value of S_N for the standard notch used is given by Equations [9] and [10] of the paper, which value is 48,000 psi. The relationship between P/A and S_N is expressed by Equation [5], which by transposition can be presented in the following form

$$S_N = \frac{P}{A} \left[1 + u \left(\frac{6e}{d - h} \right) \right] \dots \dots \dots [24]$$

For centric loading, eccentricity of loading e is 0, in which case S_N equals P/A , or 48,000 psi. This value is actually higher and not lower than the nominal breaking strengths observed in centrally notched plates, referred to by Jonassen. In the author's opinion, it is not possible to make a direct comparison of the nominal breaking strengths of steels obtained by the cleavage-tear and centrally notched coupon tests, because the notches used, the eccentricities involved, and the fractures obtained are considerably different in the two tests.

If we assume that in service conditions the worst notch is a small cleavage crack of a length of about $6/32$ in., the value of S_N of a section of steel containing such a crack is of the order of about 24,000 psi, as has been discussed in the paper. In order to verify the latter value of S_N it would be necessary to produce in D-type coupons cracks of this type, which at the same time would have the requisite orientation and depth. This would be a difficult task to accomplish. As an approximation thereto, a series of fatigue tests was made on steels 17 and 23. In these tests rotating-beam type of test coupons of circular cross section were used. In this connection it should be mentioned that the author has called attention in the paper to the fact that fatigue failures can be considered as initiating from small cleavage fractures or cracks, which are produced by repetitions of stresses.

The results of the fatigue tests referred to are presented in Table 20 and Fig. 45. It is to be noted that the endurance limit for steel 17, which is representative of the steel used for deriving Equation [15] given in the paper, is approximately 23,400 psi. The endurance limit for steel 23 is somewhat lower, it being in the neighborhood of about 22,300 psi. The results of these fatigue tests seem then to verify the extrapolated value of about 24,000 psi as being the nominal strength of a steel of a 60,000-psi tensile strength, in the presence of small cracks. Some ships have reportedly fractured under nominal stresses of approximately 24,000 psi. This appears to indicate that the mechanism of fracture under multiaxial stress systems is essentially the same, regardless of how these stress systems are created.

The third point discussed by Jonassen is related to the effect on breaking strengths of the cold work present in the pressed notch used. Quantitative data on this were obtained on several different steels in the early stages of this investigation. Of these, the P/A values and transition-temperature determinations obtained on steel 23 with milled and machined notches of approximately the same R value are presented in Table 21 and Fig. 46. It is to be observed from these data that no appreciable difference exists between the breaking loads and the transition temperatures determined by cold-pressed, and machined notches. The small differences found in the P/A values may be attributed to the slight difference in the root radii of the notches prepared by the two methods. It is apparent then that the shallow cold work produced by the pressing of the notch does not affect the test results materially. It is obvious that even in machined notches some cold-working or deformation occurs at the apex of the notch before the actual fracture initiates, the extent of which deformation is determined by the radius of the notch. As stated in the paper, in cleavage-tear tests the P/A values are determined by the entire section subjected to fracturing, and they are relatively uninfluenced by local cold-working, irregularities, or nonhomogeneity of the metal. Aging of the pressed notches was likewise found not to affect the P/A values. If the entire section of the test coupon were cold-worked, appreciably greater loads would be required to cause fracturing.

N. A. Kahn and E. A. Imbembo describe their testing procedure for determining transition temperatures by means of load-extension diagrams and fracture examinations, and comment on the sizes of the test coupons used by the author. The test coupons of various dimensions shown in Fig. 1 were used for deter-

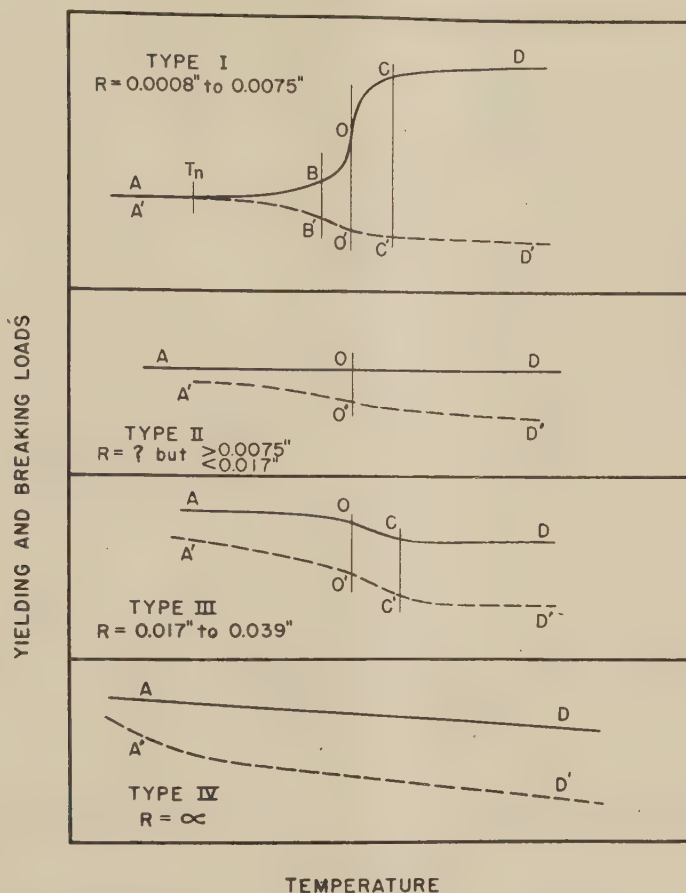


FIG. 43 FRACTURE CURVES CORRESPONDING TO VARIOUS NOTCH ROOT RADII

(Solid lines = P/A , dotted lines = Y/A , transition temperature = $O-O'$.)

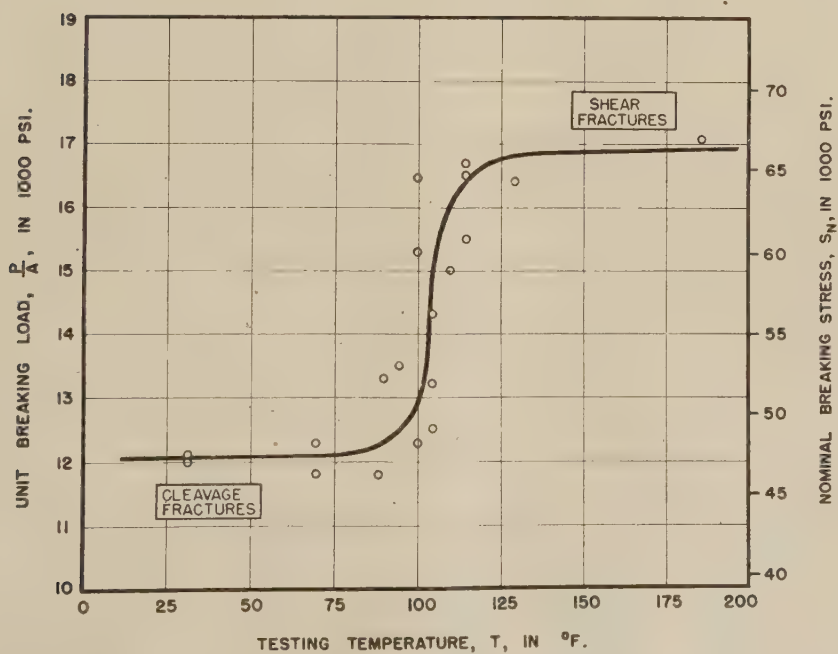


FIG. 44 TRANSITION CURVE OF STEEL 18, WITH RESULTS OF INDIVIDUAL TESTS PLOTTED TO SHOW DEGREE OF SCATTER AT TRANSITION RANGE

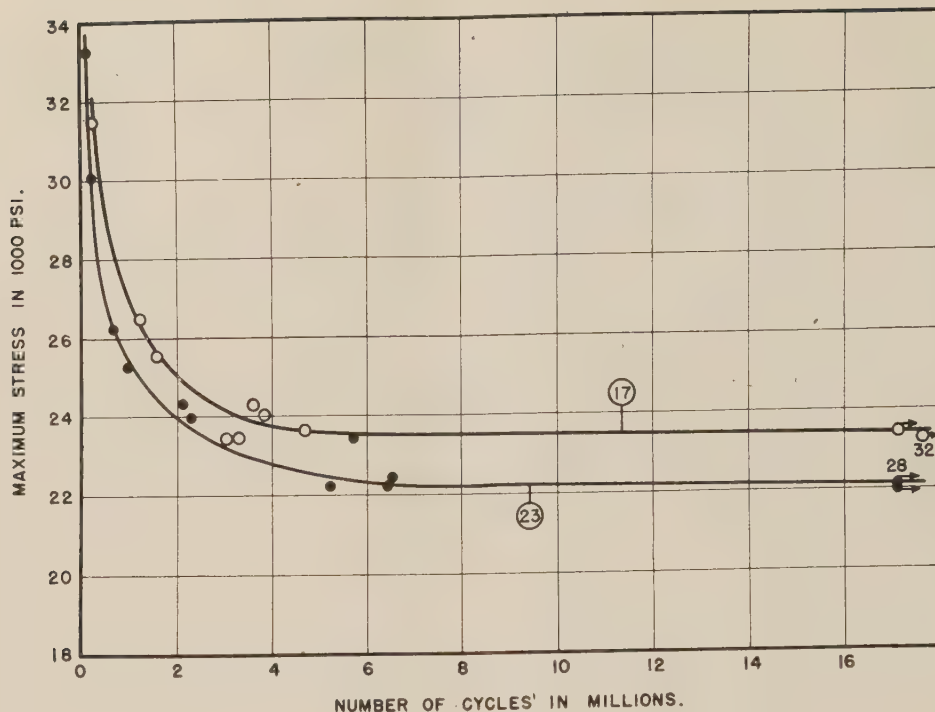


FIG. 45 FATIGUE TESTS ON STEELS 17 AND 23 FOR ESTABLISHING THEIR ENDURANCE LIMITS

TABLE 20 FATIGUE TESTS ON STEELS 17 AND 23

(Fracture transverse to rolling direction of plate)

Steel No. 17		Steel No. 23	
Maximum induced stress, psi	No. of cycles to failure, in millions	Maximum induced stress, psi	No. of cycles to failure, in millions
31400	0.214	33220	0.131
26500	1.298	30400	0.264
25500	1.670	26220	0.733
23400	3.033	25230	0.920
23400	3.389	24370	2.140
24150	3.693	23980	2.346
24000	3.838	22200	5.224
23600	4.758	23420	5.742
23400 ^a	28.384	22200	6.482
23200 ^a	32.067	22420	6.630
...	...	22100 ^a	28.030
...	...	22010 ^a	28.800

^a No fracture.

mining the effect of mass of the section; the geometry of notch on breaking and yield loads, and on transition temperatures; also to investigate the directional properties and other important characteristics of steel plates of pressure vessel and ship-plate qualities. They were not intended for determining transition temperatures only. The test coupon used by the author as standard in most tests was the D-type coupon shown in Figs. 1 and 37. Its dimensional details were developed by extensive tests described in the paper, and not chosen arbitrarily. It is apparent that the preparation, notching, and testing of the D-type coupon are simple. As stated in an earlier section of this

TABLE 21 COMPARISON OF UNIT BREAKING LOADS OF STEEL 23 AT VARIOUS TEMPERATURES, WITH PRESSED, AND MILLED NOTCHES

(D-Type coupon, 45-deg notch, of $\frac{3}{16}$ in. depth, with root radii designated) Unit Breaking Loads, P/A, psi

Test temp, deg F	Pressed Notch— Notch radius = 0.0014 in. to 0.0016 in.		Milled Notch— Notch radius = 0.0013 in. to 0.0017 in.	
	Range	Average	Range	Average
32	12300-13300	12850	12400-13800	12900
90	12600-12700	12650	13400-13600	13500
100	12700-14200	13300	14000-14700	14350
110	13500-17200	15350	14200-16200	15200
120	16000-17600	16800	14500-16800	15650
130	15000-17600	16300	16700-16700	16700
140	16000-17600	16800	16400-16500	16450
200	16000-17000	16850	16100	16100

closure, the sole determination of transition temperatures or examination of fractures does not suffice to define the quality or the behavior of a given steel under multiaxial stress conditions and at low temperatures.

If it were desired to determine transition temperatures alone, use could be made of the beam or DX-type test coupon shown in Fig. 35. As stated in the paper, coupons of this type containing notches of different sharpnesses were used successfully for surveying the effect of notch root radii on the extent of deflection obtainable. For comparable results, it is necessary to adjust the span of this beam coupon so that it will have essentially the same relative rigidity as the D-type coupon. The dimensions shown in Fig. 35 accomplish this, and were found to be satisfactory for testing plates within the thickness range of $\frac{1}{2}$ to 1 in., inclusive. The DX-type coupon is undoubtedly more economical than the Charpy coupon, or that proposed by Kahn and Imbombo. Its size is only 3 in. $\times$ 9 in., the full thickness of the plate to be tested being its thickness. As has been stated previously, this test coupon can be cut from the ends of duplicate tensile- or bend-test blanks furnished by steel mills, and therefore requires no extra expense. Its notching by the pressing method is likewise very simple.

In Fig. 47 are shown the transition curves of steel 18 determined by DX-type and D-type coupons. The breaking loads obtained at various temperatures by the use of the DX-type coupons are given in Table 22, and the P/A values obtained by the use of the D-type coupons are given in Table 16. For DX-type

TABLE 22 DETERMINATION OF TRANSITION-TEMPERATURE RANGE OF STEEL 18 BY USE OF DX-TYPE TEST COUPON

(Pressed notch: 45 deg, of $\frac{3}{16}$ in. depth, with 0.0015 in. root radius)

Test temp, deg F	Breaking load, lb	Fracture
50	64000-65000	Cleavage
75	64000-66000	Cleavage
90	64000-65000	Cleavage
100	67000-73000	Cleavage
110	101000-103000	Shear
120	100000-102000	Shear
130	102000-103000	Shear
140	102000-104000	Shear
150	100000-102000	Shear

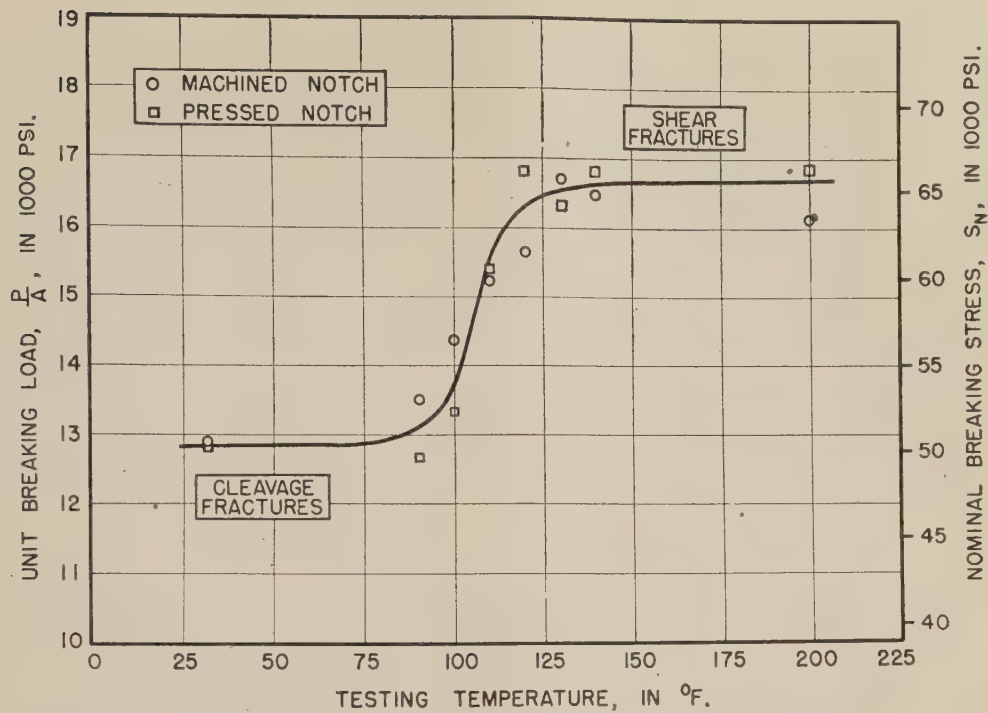


FIG. 46 DETERMINATION OF TRANSITION CURVE OF STEEL 23 BY USE OF MACHINED AND PRESSED NOTCHES (No appreciable difference was found in P/A and T_0 by use of these two methods of notching.)

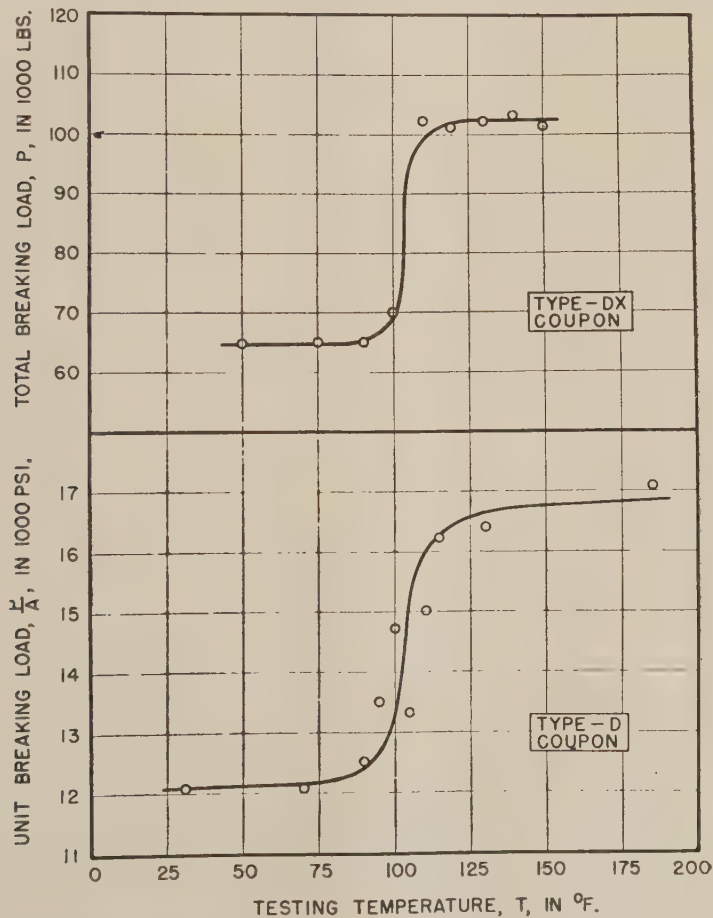


FIG. 47 TRANSITION CURVE OF STEEL 18 DETERMINED BY TYPES D AND DX COUPONS, SHOWING SAME TRANSITION TEMPERATURE

coupons the total breaking loads, and for the D-type coupons the average values of P/A were plotted against the testing temperature. In both cases the notch used was 45 deg, $3/16$ in. deep, with root radius of 0.0015 in. It is to be observed that the transition temperature of this steel determined by these two types of test coupon is approximately the same. It is apparent then, that by the use of this simple coupon it is possible to determine the transition-temperature range of steels by measuring the breaking loads at various temperatures, without resorting to load-extension diagrams or to measurement of energy of crack propagation.

The fractures shown in Fig. 23 reveal a marked reduction of area at the notch, indicating that the initial fractures originating from the notch used by Kahn and Imbembo ($R = 0.039$ in.), are in every case of the shear type. Their load-extension diagrams given in Fig. 24 reveal the same situation. It is probably for this reason that the load required to cause fracturing of their test coupon remains practically unchanged at the temperatures reported. This is shown to be true for notches of 0.039 in. root radius by the data presented in Table 18 and in Figs. 39 and 40, and would be expected to be the case according to type III diagram shown in Fig. 43. The data cited indicate that Kahn and Imbembo would have obtained an appreciable increase in the breaking loads had they conducted tests at temperatures somewhat lower than the range they have covered. The D-type coupon used by the author has a larger cross section and contains a notch of much smaller root radius ($R = 0.0015$ in.), by virtue of which the notch geometry remains one of the main determinants of the load level at which yielding and fracturing occur. In the author's tests, the load-carrying capacity, transition temperatures, and other characteristics are revealed by the observed differences in the breaking loads.

Kahn and Imbembo inquire about the work diagram obtained in cleavage-tear tests. Fig. 50 contains two such diagrams, one being typical of the cleavage type, and the second typical of the shear type of fractures. It is to be noted that the diagrams for these two types of fractures are distinctly different from each other. The diagram for the fracture at 32 F is strictly of cleavage type, in which the fracture occurred while the steel was, practically speaking, still within its elastic range. On the other hand, the diagram for the fracture at 130 F shows a distinct yield point at a lower load, and an ultimate rupture at an appreciably higher load than the former fracture. In the cleavage fracture at 32 F the work diagram terminates abruptly as soon as fracturing is initiated, since the coupon breaks into two pieces, whereas for the fracture at 130 F the diagram extends beyond the initiation of fracturing.

The statement made in the paper regarding the use of certain ratios of eccentricity to coupon depths was intended to call attention to the possibility of obtaining in the test-piece tension without bending. Actually, for the study of cleavage fractures, the use of coupons of the latter design offers no advantages. As has been stated, fractures of strictly cleavage type occur mostly at about the upper edge of the elastic range and propagate very rapidly, producing no appreciable permanent deformation. Under the circumstances, the neutral axis does not remain stationary, but shifts as the fracture extends itself.

Kahn and Imbembo also inquire about the part a given steel plays in determining its mode of fracturing with the use of different notches and at a given temperature. Inasmuch as the transition temperatures of different steels determined by the same notch is not the same for all steels, it is obvious that the steel itself establishes its own transition temperature for the given notch and test conditions. This is expressed concisely by Equations [22] and [23]. The slope of the T_g versus R curve shown in Fig. 41 is the characteristic of the steel, since it represents the

response of the steel to the stress systems created by notches of different root radii.

Reference is made by A. B. Kinzel to the test coupon used some time ago by E. G. Coker.³⁴ Professor Coker's coupon is essentially a 2-in.-diam disk, which is milled into the shape of the letter "C" in order to obtain a recess or slot for accommodating the frame type of levers used, the loading being applied by means of the levers. It appears that the stresses induced in this test coupon are similar to those encountered in a "guided-bend" test coupon, since the stress distribution is modified by the levers. The notch used by Coker is a $1/8$ -in.-diam drilled hole, which is not of sufficient acuity to be revealing. In the author's coupon the loading is direct and of the hinged type, by virtue of which no similar complex stresses are introduced, and special care is taken to keep the notched edge of the coupon in a "free-bend" condition, such as is obtained in a simple beam supported at the ends and loaded in the center by a concentrated load. Also, much sharper notches are used in the latter coupons. These features are essential for rendering the testing procedure sensitive, and the applied stresses less complicated.

The heat-affected zone near the weld, and details of weldability of steels are undoubtedly important, and the author agrees that they should be studied. The present investigation was confined primarily to the study of the behavior of steel plates and butt welds under various stress conditions, and of their susceptibility to cleavage-fracturing. From this base, the investigation could be extended to cover welding and weldability. It is important to investigate the behavior of steel plates, since the damage caused by, and the extent of propagation of, cracks are largely determined by the properties of the structural steel plates, regardless of how or where the cracks initiate.

The rate of loading, within the limits investigated, was found to be ineffective to produce any noticeable change in breaking loads. It appears that once cleavage cracks have initiated, the rate of loading becomes inconsequential, since it is difficult, if not impossible, to overtake cleavage-crack propagation by any loading rates available in ordinary testing machines. Theoretical and experimental data seem to indicate that cleavage cracks propagate in a manner similar to stress waves, in which case their rate of propagation would approach the velocity of sound in the material.

The effect of temperature on breaking loads of several representative steels, shown graphically in Fig. 15 of the paper, requires no detail elaboration. It is believed that Table 11 contains sufficient data to establish the effect of temperature at the transition range, which is evident by the additional data presented in Tables 16 and 18 of this closure. For the sake of legibility, all the test data given in the paper and in Table 11 were not plotted in Fig. 15. For example, the behavior of steel 15 over a wider range of temperature than that shown in Fig. 15, could be obtained by plotting the test data given in Table 11 for this steel, in the as-received condition, for 45- and 60-deg notches, as shown in Fig. 48. This curve covers the temperature range of 10 F to 200 F, and was constructed by making allowance for the fact that the breaking loads for 60-deg notches are slightly higher than those for 45-deg notches. The broader range of temperature covered by Fig. 48 reveals nothing significant other than the abrupt change in the P/A values at the transition temperature, which change is likewise shown in Fig. 15 by the curve of the steel.

The transition of steel from the cleavage to the shear type of fracturing, as effected by the change of temperature, can be represented by the idealized diagram included in Fig. 49. In this diagram the unit breaking load P/A , unit yield load Y/A , and the

³⁴ "An Experimental Investigation of Cracking in Mild-Steel Plates and Welded Seams," by E. C. Coker and B. P. Haigh, *Engineering*, London, England, vol. 139, May 3, 1935, pp. 476-478.

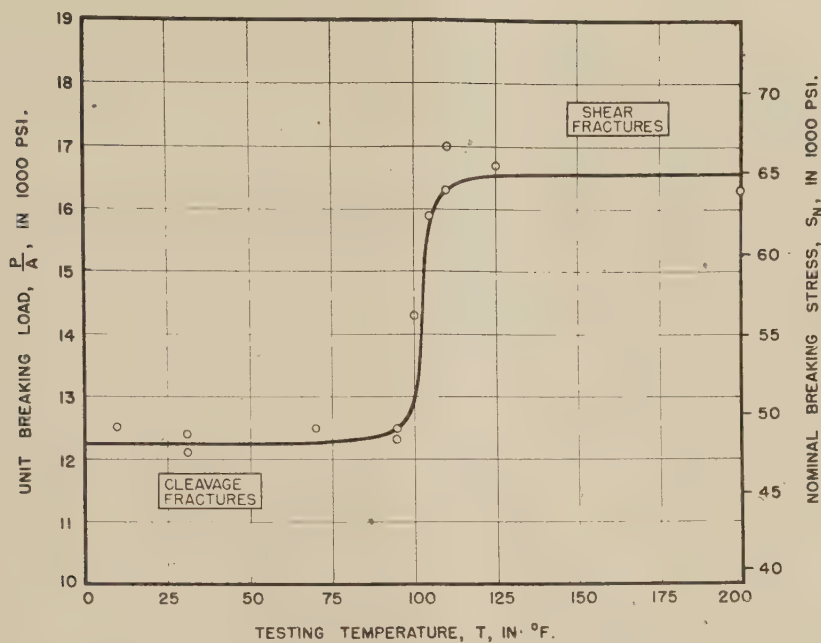


FIG. 48 TRANSITION CURVE OF STEEL 15, COVERING TEMPERATURE RANGE 10-200 F

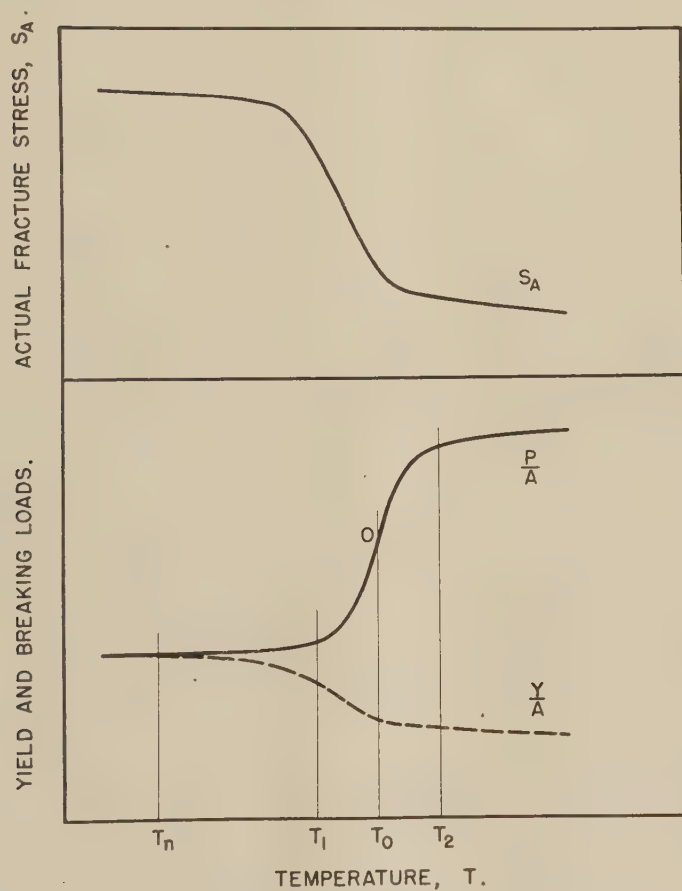


FIG. 49 CLEAVAGE-TEAR-TEST DIAGRAM SHOWING TRANSITION FROM CLEAVAGE TO SHEAR TYPE OF FRACTURING BY CHANGE OF TEMPERATURE

actual unit stress S_A , are plotted against the temperature. It is to be noted that these values change abruptly within the temperature range of $T_1 - T_2$. The change in the P/A and S_A values is undoubtedly in sympathy with the abrupt change which occurs in the Y/A values, occasioned by the presence of the notch. It is known that the critical yield stress increases with decrease of temperature, which increase is, however, a gradual one in sections containing no notch.³³ It follows that the change observed by the cleavage-tear tests in the P/A and Y/A values is caused by the presence of the notch. It appears then that the K versus T curve, K being the stress-concentration factor, is approximately parallel to the Y/A versus T curve, since the degree of stress concentration increases with the increase of the yield stress and decreases with the decrease of the yield stress.

The curves in Fig. 49 are assumed to be symmetrical with respect to their respective points of inflection, with positive and negative slopes, this point for the P/A curve being the point O . With this assumption, it can be seen that the equation for these curves is of the following form

$$\Delta S = a(\Delta T)^b \dots \dots \dots [25]$$

where ΔS is the change in the unit load or stress values, as the case may be, corresponding to the temperature change of ΔT , and a and b are constants for each steel in a given condition.

For expressing the P/A values as a function of T through the transition range, Equation [25] can be written in the following form

$$\frac{P}{A} = \left(\frac{P}{A}\right)_0 + a(T - T_0)^b \dots \dots \dots [26]$$

where

(P/A) = unit breaking load at temperature T

$(P/A)_0$ = unit breaking load at temperature T_0

T_0 = temperature for point of inflection, or theoretical transition temperature.

a and b = constants for a given steel and test conditions

For the steels tested, the exponent b of the temperature term in Equation [26] is a rational number and quotient of two odd integers, according to experimental data, so that the term $(T - T_0)^b$ yields a real and negative value if $T < T_0$. Otherwise, Equation [26] would apply only to cases where $T > T_0$, the corresponding equation for $T < T_0$ being

$$\frac{P}{A} = \left(\frac{P}{A}\right)_0 - a(T_0 - T)^b \dots \dots \dots [26a]$$

On the basis of Equation [26] and by the use of the test data given in Tables 11 and 16, equations were derived for the several steels tested for expressing their transition from the cleavage to shear mode of fracturing, as a function of change of temperature. For D-type coupons containing the standard notch these equations are as follows

Rimmed Steels

$$\text{Steel 7 (1 in. thick)} \dots \dots \frac{P}{A} = 15,600 + 980(T - 107)^{1/3} \dots [27]$$

$$\text{Steel 22 (1/2 in. thick)} \dots \dots \frac{P}{A} = 14,150 + 700(T - 92)^{1/7} \dots [28]$$

Semikilled Steels

$$\text{Steel 15 (1 in. thick)} \dots \dots \frac{P}{A} = 14,400 + 1300(T - 103)^{1/7} \dots [29]$$

$$\text{Steel 17 (1 in. thick)} \dots \dots \frac{P}{A} = 14,000 + 1300(T - 83)^{1/7} \dots [30]$$

$$\text{Steel 18 (1 in. thick)} \dots \dots \frac{P}{A} = 14,300 + 1300(T - 165)^{1/7} \dots [31]$$

$$\text{Steel 23 (15/16 in. thick)} \dots \dots \frac{P}{A} = 14,800 + 1300(T - 110)^{1/7} \dots [32]$$

Killed Steels

$$\text{Steel 16 (1 in. thick, Ti-treated)} \dots \dots \frac{P}{A} = 16,400 + 440(T - 68)^{3/7} \dots [33]$$

$$\text{Steel 19 (1 in. thick, Si-killed)} \dots \dots \frac{P}{A} = 16,500 + 310(T - 65)^{3/7} \dots [34]$$

$$\text{Steel 20 (2 1/4 in. thick, Si-killed)} \dots \dots \frac{P}{A} = 14,600 + 1450(T - 120)^{1/7} \dots [35]$$

The breaking loads computed by the foregoing equations agree reasonably well with the corresponding experimentally determined values, indicating that the transition curves of all these steels are of the same general form as the curve in Fig. 49, and that the cleavage-tear transition curves in Fig. 15 illustrate satisfactorily the P/A versus T relationships. Furthermore, it is to be noted that in these equations the constants and the exponents are different for the different grades of steel of the same thickness. The effect of thickness on the behavior of steels of the same grade through the transition range is clearly revealed by the equations. For example, in Equation [28] which covers a rimmed steel of 1/2-in. thickness, the exponent for the temperature term is 1/7, whereas for the 1-in.-thick steel of the same grade covered by Equation [27], it is 1/3, indicating that 1/2-in.-thick rimmed steel behaves more like the 1-in.-thick semikilled steels. Similarly, the 2 1/4-in.-thick silicon-killed plate exhibits transition characteristics of 1-in.-thick semikilled steel, as a comparison of Equations [31] and [35] shows. These observations indicate that the testing method used is sensitive and offers possibilities of classifying steels of different grades with respect to their behavior through the transition range.

Kinzel states that the stress conditions created by the notch used are too severe to be representative of actual failures in service. According to the data summarized in Table 19, the transition temperatures of our present structural steels in the as-received condition are considerably above atmospheric temperatures for notches with root radii of about 0.0015 in. or smaller. Jonassen quotes data in his discussion of this paper to the effect that for some steels the transition temperatures determined by wide-plate tests have been about 90 deg F higher than that indicated by the standard Charpy test. DeGarmo's tests³⁵ on hatch-corner models have likewise shown transition temperatures for these structures of about 86 deg F higher than those determined by the Charpy impact test. Fig. 30 submitted by Peterson shows an actual ship failure by fracturing at about 75 F. It appears then that the transition temperatures indicated by the cleavage-tear tests are fairly representative of failures observed in service.

It is for the reasons given in the preceding paragraph that notches of smaller root radii were used in this investigation, the object being to approach stress conditions which often prevail under service conditions. It seems reasonable to assume that the controlling stress condition in monolithic or massive structures in service are those created by a small cleavage crack³¹ since such a notch is one of the nearest approaches to triaxiality. What other stress conditions or notch geometry could be logically justified by the present data? This theory cannot be considered invalidated by the observation that the number of failures by fracturing increases with decrease of temperature. There are

³⁵ Some Tests of Large Welded Structures," by E. P. DeGarmo, J. L. Meriam, and R. C. Grassie, *The Welding Journal*, vol. 26, 1947, pp. 257-267.

other basic reasons for the increase of failures at low temperatures, which have already been discussed and will be further elaborated upon in this closure.

The arrest of the cleavage fracture by an interposing shear fracture illustrated in Figs. 18(a) and 18(b) of the paper, is discussed by Kinzel, in which connection he infers that such an arrest is due to the metallurgical properties of the steel. The author's observations of numerous cracked plates from failed ships and of fractured test pieces indicate that major cleavage cracks show very little, if any, permanent deformation, and that such cracks terminate almost entirely by shear fractures characterized by considerable plastic deformation. This change in the fracture type cannot be explained by metallurgical properties, since, if the steel were capable of resisting cleavage-fracturing, it would not have fractured in that manner initially. It is more likely to be due to the change in the state of stress brought about by the progress of the crack, since the propagation of cleavage cracks relieves potential energy and impairs the rigidity of the structure. It is probable that as a certain limit of depth of penetration is reached, insufficient potential energy and rigidity remain in the structural system to support further progress of the cleavage crack; thereby some flexing occurs to convert the fracture to the shear type. But since greater loads are required to continue shearing, the crack is arrested at that point unless loads exceeding the maximum shear stress were present. There appear to be no valid reasons for assuming that the contour of, and the stress conditions at the terminus of a shear crack are the same as those at the terminus of a cleavage crack.

Undoubtedly, Kinzel will agree that the ductility of a metal or its ability to undergo permanent deformation is not an independent property. It is a property dependent on several factors, including the state and the manner of application of stress, temperature, and the inherent metallurgical characteristics of the metal. Therefore it cannot be defined without defining these factors. The load-carrying capacity of the metal is likewise dependent on the factors mentioned. The cleavage-tear-test data presented in the paper and in this closure show that a direct relationship exists between the load-carrying capacity, or P/A values, and the behaviorism, toughness, or ductility of the metal under given conditions. Therefore in these tests the behaviorism or ductility of the metal is expressed in the term of unit breaking loads. Inasmuch as it is impossible to have ductility without yielding, it is important to note the relationship of yield loads to breaking loads and to the actual fracture stress shown in Fig. 49, in order to visualize this point. It is clearly evident from this figure that a correlation exists between the curves representing the foregoing values, with respect to changes in their respective slopes. As Y/A increases by decrease of temperature, P/A decreases, and S_A increases by the resultant increase in the stress-concentration factor, and vice versa. Accordingly, in cleavage-tear tests, the ductility and behaviorism are reflected by the observed changes in the P/A values. This is substantiated by the P/A and Y/A curves shown in Fig. 42.

Further experimental confirmation of the statements given in the preceding paragraph is presented by the breaking load versus time charts included in Fig. 50, which were obtained on steel 7 above and below its transition temperature by the use of the D-type coupon con-

taining the standard notch. These diagrams were constructed from data given in Table 23, which were obtained by photographing by a motion-picture camera the load-indicating dial of the tensile testing machines simultaneously with a running clock, as the test piece was loaded to fracture. By this procedure the effective rate of loading was established for the entire test coupon as a unit. The characteristic variations in the rate of loading of the testing machine, as affected by the level of loads reached, was determined by a separate calibration test. The corresponding corrections were then applied to the observed rate of loading. With these corrections, the elapsed time becomes directly proportional to the deformation occurring in the

TABLE 23 DETERMINATION OF EFFECTIVE RATE OF LOADING AND OF MODE OF FRACTURING, DATA OBTAINED CINEMATOGRAPHICALLY

Applied tension, lb	Temperature of test, 32 F		Temperature of test, 130 F	
	Elapsed time, sec		Elapsed time, sec	
	Observed	Corrected	Observed	Corrected
5000	Start	...	Start	...
10000	1.5	1.0	1.2	0.8
15000	2.2	1.4	2.0	1.3
20000	3.0	2.0	2.7	1.8
25000	3.6	2.4	3.2	2.1
30000	4.0	2.8	3.8	2.7
35000	4.6	3.3	4.2	3.0
40000	5.2	3.8	4.7	3.4
45000	5.8	4.4	5.0	3.7
50000	6.0	4.5	5.4	4.0
55000	6.8	5.3	5.9	4.5
60000	7.2	5.7	6.2	4.8
62000	...	...	6.6	5.2
65000	7.6	6.1	6.8	5.4
67500	...	...	8.0	6.6
70000	8.0	6.5	8.8	7.4
75000	8.4	6.9	10.2	8.8
78000	9.0	7.5	...	...
78000	10.0	8.5 ^a	...	...
80000	...	...	11.7	10.3
85000	...	...	13.7	12.3
90000	...	...	16.3	14.9
95000	...	...	20.0	18.6
97000	...	...	25.0	23.6 ^b

^a Cleavage fracture.
^b Shear fracture.

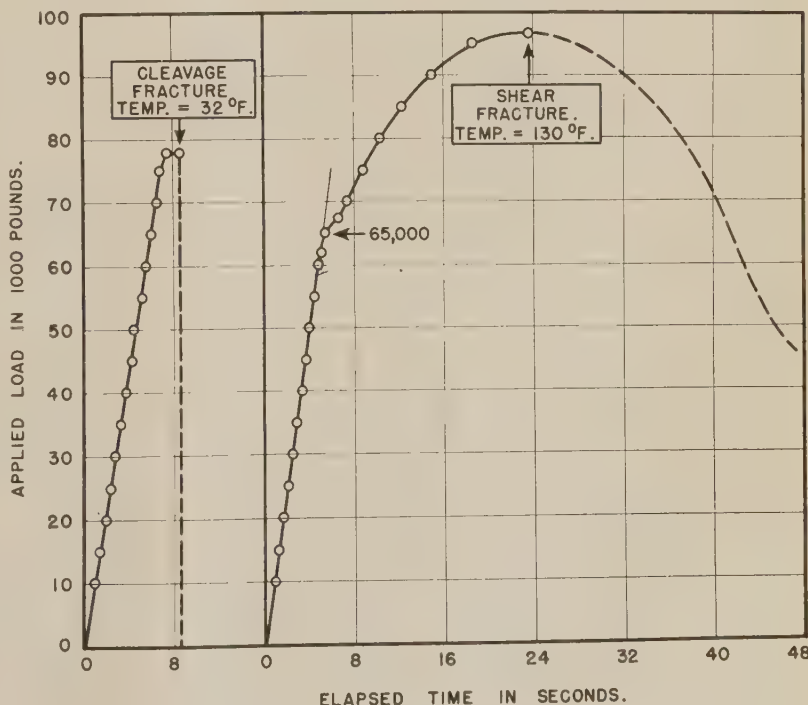


FIG. 50 LOAD OR WORK DIAGRAM OF FRACTURING OF STEEL 7 AT 32 F, CLEAVAGE FRACTURE, AND AT 130 F, SHEAR FRACTURE

(Standard D-type coupon containing standard notch used in both cases.)

test piece. Therefore the charts in Fig. 50 can be considered as being the work diagrams of fracturing.

Fig. 50 shows that no yielding or appreciable plastic deformation is apparent at 32 F, and that a sudden rupture of the entire test coupon occurs at this temperature at a tensile load of 78,000 lb. This is a typically cleavage type of fracture. At 130 F a definite yielding is observable at a load of about 65,000 lb beyond which an appreciable plastic deformation occurs until a load of 97,000 lb is reached, at which level the steel fractures by shearing or slow separation, exhibiting good ductility. This is typical of the ductile or shear fracture. Within these two extremes there are fractures which show intermediate ductility, the extent of ductility decreasing with the decrease of temperature, as shown in Fig. 42. The significance of these observations with respect to service performance has already been discussed in connection with the treatment of Enzian's comments.

It is apparent then that the behaviorism of steels is clearly integrated by the load required to cause fracturing under the test conditions used. A similar integration would be extremely complicated, if not impossible, in terms of elongation or reduction-of-area measurements, since the latter vary immensely, depending upon where the measurement is taken. If this measurement of plastic deformation or reduction of area were made at or a little below the notch, no significant conclusions could be reached, since a steel showing evidence of ductility at or near the notch might break almost like cast iron below the notch, exhibiting no noticeable plastic deformation. The P/A values, on the other hand, show the behavior of the entire section of the metal, "as a unit," and therefore serve as a better index for evaluating the behaviorism or toughness of a structure. It is obvious from this that the P/A values were measured in this investigation not only with the object of establishing the nominal breaking strengths, but also for effectively and correctly evaluating the behaviorism of a given steel under multiaxial stress conditions.

Most massive and monolithic structures which fail in service do fail by cleavage type of fracturing, unless the stresses involved under service conditions are essentially uniaxial and no discontinuities are present. It is clear from Fig. 50 that at low temperatures, and under multiaxial stress conditions, it is possible for the yield load almost to coincide with the breaking load, and that a considerably lower load will suffice to cause cleavage type of fracturing than the load required to produce the shear or ductile type of fracturing. It is also apparent from the data presented in Table 19 that the transition temperatures of the present structural mild steels are considerably above atmospheric temperatures if multiaxial stress conditions prevail. Numerous monolithic structures, such as all-welded ships and pressure vessels, made of as-rolled mild steels have fractured in the brittle manner under a nominal stress of about 24,000 psi, as predicted by Equation [16] given in the paper. However, as has been stated in connection with Bennett's discussion in the earlier section of this closure, failures of this type occur only if several independent conditions co-exist or prevail simultaneously; they are not due to a single cause. This appears to explain why only relatively few ships and pressure vessels have failed by cleavage-fracturing, and others made of seemingly the same steel and design have not failed. Nevertheless, unless other explanations could be found for such failures, it would appear unsound to assume that those which have not failed should not fail under similar conditions. Failure by cleavage-fracturing is more likely to occur in all-welded tanker ships and large pressure vessels because the probability of having all the necessary conditions simultaneously present for failure in such structures is greater, due to the complexity of their service and stress conditions.

Mr. H. G. Larew comments on the desirability of photoelastic studies of the stress patterns involved in cleavage-tear test cou-

pons and discusses the effect on the breaking loads of the variations of the 4-in-wide slot used in the coupons, Fig. 1. The author agrees that considerable knowledge could be gained by photoelastic studies. However, actual tests made in the early stages of this investigation have shown that variations in the slot width within the range of 2 to 16 in. have no appreciable effect on the breaking loads. The results of these tests are summarized in Table 24.

TABLE 24 EFFECT ON BREAKING LOADS OF VARIATIONS IN SLOT WIDTH OF CLEAVAGE-TEAR TEST COUPON

Modified D-type coupon containing 45-deg notch of $\frac{3}{16}$ in. depth and 0.0008 in. root radius

Slot width, in.	Unit breaking load, P/A , psi—	
	Range	Mean
2	11400-12200	11800
4 ^a	11300-11600	11450
8	11100-11300	11200
12	11200-11500	11350
16	11400-11700	11550
	Average.....	11470

^a D-Type Coupon, Figure 1.

NOTE: With the exception of the slot width and dimension c , all other dimensions were the same as those for coupon D-Type, Fig. 1. Dimension c was increased or decreased by the same length as the slot width was increased or decreased.

Mr. Larew's second comment deals with the factor n , used in Equation [3] of the paper. In order to explain the procedure followed, the latter equation can be written in the following form

$$s = a \left(\frac{P}{A} \right) + qb \left(\frac{Mc}{I} \right) \dots \dots \dots [36]$$

where a and b are the correction factors or the relative stress-concentration factors for the axial tension and the flexural terms, respectively, introduction of which is necessitated because of the presence of the notch. By dividing Equation [36] by a there is obtained

$$\frac{s}{a} = \frac{P}{A} + q \left(\frac{b}{a} \right) \left(\frac{Mc}{I} \right) \dots \dots \dots [37]$$

In the absence of eccentricity of loading, the flexural term is zero, in which case $s/a = P/A = S_N$, since by definition S_N is the nominal breaking stress for centric loading. The term b/a is the ratio of the relative stress-concentration factors, which is designated by the symbol n . Substitution of S_N for s/a , and n for b/a in Equation [37] yields Equation [3].

As stated in the paper, the numerical values of n were computed from the P/A versus eccentricity of loading determinations summarized in Table 4. The stress-concentration factors are different for different notch and section geometries, but their ratio appears to remain constant. Therefore it is obvious that the use of the factor n introduces no errors. It does not appear that the introduction of variable correction factors of unknown numerical values would simplify Equation [10] of the paper.

E. M. McCutcheon discusses crack arresters as applied to all-welded ships. He also furnishes an analysis of the P/A versus d curve presented in Fig. 8 of the paper. The author agrees that the insertion of riveted straps or joints at the critical sections of ships as "crack arresters," was undoubtedly one of the most expedient methods that could be applied under war-emergency conditions. However, its perpetuation as a permanent feature and as the best assurance against crack propagation in welded structures appears to be objectionable on several grounds. The following are some of the objections: (a) Due to the structural heterogeneity created by insertion of riveted seams in a monolithic structure, these seams are likely to invite stresses and flexures greatly in excess of their proportionate share obtained in a structure of all-riveted construction. (b) The rate of corrosion and loosening of rivets will undoubtedly be accelerated by the exces-

sive flexing and stressing referred to in (a), and thereby serious leakage may develop, which would be particularly troublesome in the straps at the bottom of the ship. (c) The general practice used for slotting the shell of the ship for installation of riveted straps consists of torch-cutting, which in most cases, instead of being a smooth cut, is irregular and may contain numerous shallow but sharp notches, from which cleavage cracks could initiate. (d) Installation of one or several riveted straps does not constitute a positive guarantee against crack propagation, since failure has also been observed in all-riveted ships.³⁶

A crack arrester should be capable of performing the two independent functions to be discussed before it could be considered as being a satisfactory safeguard for the structural safety of an all-welded ship, made of a steel which is susceptible to cleavage fracturing at temperatures and under loading conditions encountered in service. The first function consists in the instantaneous halt or arrest of cleavage cracks once they reach the crack arrester. The second function consists in assuring that no new cracks will initiate from these arrest points, even after the structure is weakened and the effect of stresses is intensified by formation of the crack. Mr. McCutcheon shows that the riveted seam or strap is capable of performing the first function. He infers that it will also perform satisfactorily the second function, evidently on the basis of his present data, which show that no cracks have continued through the riveted seams. However, the assurance offered by the riveted seam against continuation of cracks cannot be considered as being positive in all cases.

The perforated plates or drilled holes proposed by the author are similar in behavior to the riveted seam; they perform satisfactorily the first function as the test data show, but unsatisfactorily the second function. Accordingly, both the perforations and the riveted seam act as crack arresters only in the sense of "shock absorbers," and their utility, by themselves, is therefore limited. If sufficient stress were present to initiate a crack in the deck or strakes of a ship, its effect may increase by fracturing, due to the resultant increase in the eccentricity of loading. The latter may be of sufficient intensity to initiate a new crack in the abutting plate edges of the riveted seam, or in the edges of one of the holes of the perforated plate. It is obvious then that the assurance against continuation of cracks is not absolute with either design, and that it is necessary to provide other safeguards in the structure, in addition to these crack arresters.

The use of greater factors of safety in design introduces such additional safeguards, since the data presented in the paper show that cleavage fractures do not initiate unless finite unit breaking loads are reached. Inasmuch as failures have occurred in all-welded ships, it appears that these load levels are actually being reached under service conditions in some of these ships. Increasing the shell thickness or using steels of higher tensile strength in the critical sections accomplishes this. Another method of increasing the factor of safety consists of installing in the ship's structure longitudinal girders or beams of sufficient strength, capable of carrying a good portion of the induced stresses, which girders can be kept separate and not built integral with the hull of the ship in order to eliminate extension of cracks into these girders. Girders of this type have actually been installed in a considerable number of T-2 tanker ships. However, in a multi-phase structure of this type the possibility exists that the hull of the ship and these strength girders may not share the load simultaneously. This tendency may be occasioned by the fact that the hull and these strength girders may not be installed with the same effective rigidity when the ship is built, and a slight but definite "slack" may be present in one of these members. A

similar situation may also arise by any differential thermal expansion or contraction of these members or of the hull.

It is apparent then that even with the foregoing modification of design it may still be possible for some cleavage cracks to be initiated in the shell of the ship. It is in view of this that the author has suggested the use of perforations or drilled holes in certain critical sections of the hull as "crack arresters." These perforations or holes should be of a staggered pattern and oriented in the same relative positions as the present riveted seams, by which arrangement more than one hole will be in the path of the crack. Such perforated plates or similar arrangements could actually be incorporated in the design of new ships with little additional expense. If used in conjunction with the design modifications mentioned, cleavage cracks should be effectively arrested at these perforations, through development of sufficient deformation and stretching at the first or second holes. Thereby, the strength girders will be engaged to share the stress and the hull will be relieved simultaneously of the excessive stresses. The use of the perforation type of crack arrester is preferred because it is free of most of the objections of the riveted seams, and is of a more economical design than the latter.

Welding of perforated straps to the unperforated shell, or of unperforated straps to the perforated shell, permits no leakage since one of these overlapping plates contains no holes. Likewise, joining of the straps to the shell by welding should cause no difficulty, if these straps were used in accordance with the arrangement described in the preceding paragraphs. Cracks are not likely to penetrate the entire width of the strap by passing through several holes in its path, since the degree of localized bending deflection required to initiate a new crack from each successive hole is not available due to the restrictions imposed by the structure of the ship. There are of course many designs other than the riveted seam and perforation type of crack arresters, discussion of which is beyond the scope of this paper. These should be considered carefully and analyzed instead of perpetuating the use of riveted straps. It may even be possible to eliminate the need of crack arresters by proper modifications of design and material specifications, in new ships.

McCutcheon's analyses of the effect of section size on the P/A values, made on the basis of the results obtained on 72-in-wide test plates, confirm the general validity of Equation [10]. Nevertheless, it appears from these analyses that the internally notched coupons, even if made as wide as 72 in., do not produce as much rigidity as is obtained in the D-type cleavage-tear test coupon. This is evidenced by the fact that in the latter coupon the maximum reduction of area obtainable, at temperatures below the transition range, is of the order of 0.01 per cent, whereas in the internally notched wide plates considerably greater percentage of area reduction occurs. This is also evidenced by the fact that, contrary to Mr. McCutcheon's prediction, no buckling was encountered in 1-in-thick cleavage-tear test coupons of depths up to and including 24 in., nor in $1/2$ -in-thick steel coupons of depths up to and including 18 in., in tests conducted at temperatures below the transition range.

It is generally conceded that riveted joints have greater damping capacity and somewhat more flexibility than the welded joint. The term "flexibility" used in the paper is in the sense of "non-rigidity." The stress-concentration factor of 9 referred to in the paper is not based upon the 60,000-psi nominal tensile strength, but on the actual fracture strength, which is estimated to be in excess of 200,000 psi. According to Equation [16], the nominal breaking strength of steel, containing a notch in the form of a $5/32$ -in-long cleavage crack, is about 24,000 psi. The ratio of these "actual" to "nominal" strengths yields a stress-concentration factor of about 9. The relationship between P/A values and actual fracture stress is illustrated in Fig. 49 of this closure.

³⁶ "Technical Progress Report of the Ship Structure Committee," *The Welding Journal*, vol. 27, 1948, Research Supplement, pp. 377-s-384-s.

The data presented in the paper show that cleavage fractures develop at a point almost within the elastic range of steel. This is illustrated by the charts included in Fig. 50, and being based on factual observation, should not be considered as a theory. The author agrees with McCutcheon that failure of ships is the exception rather than the rule, which subject has been discussed somewhat in length in connection with the treatment of Bennett's and Enzian's comments, and need not be repeated here.

R. E. Peterson includes an interesting chart, Fig. 30, in his discussion, in which the transition curves of steels 7 and 16 are compared with the curve of ship-service versus temperature, and states that not much change is indicated by the transition curves to account for more ship failures at low temperatures. The explanation of this apparent discrepancy can be found by referring to Figs. 42, 43, 49, and 50, of this closure. It is to be noted in these figures that significant changes occur in steels at temperatures below the transition range, which is manifested by the increase in the load required to cause yielding and by the decrease in the load required to cause fracturing. It is apparent that the extent of yielding decreases with decrease of temperature and becomes practically zero at the temperature T_n , shown in Fig. 42. As explained in connection with the discussions of Enzian and Kinzel, some yielding occurs in steels between the temperature T_n and the transition temperature T_0 , which yielding may be sufficient to reduce the stress peaks and thereby in some cases prevent fracturing. Therefore the slope of the yielding and fracturing curves, or $d(Y/A)/dT$ and $d(P/A)/dT$, below the transition temperature range, are good indexes of the increasing tendency of steel to fracture by the brittle mode with decrease of temperature.

The author agrees with Peterson that temperature differences can induce high stresses. There are cases on record showing that oil storage tanks and tanker ships have ruptured due to the added stresses induced by undue thermal fluctuations or by sudden climatic changes. Design modifications have been found necessary in ships containing large refrigeration compartments or sections, in order to eliminate cracking of the structure by the stresses imposed by the temperature differences. Thermal stresses are additive to the static stresses and therefore may increase the danger of fracturing.

R. W. Roop comments that the use of the unit-breaking-load values as a discriminant of toughness and other characteristics of steels was decided upon on the basis of a small number of tests. It is stated in the paper that the first "indication" of P/A being such a discriminant was obtained in the simple beam tests. Actually, the usefulness of P/A was demonstrated by a large number of tests presented in numerous figures and tables of the paper. This has been discussed in detail in this closure, in connection with the comments of several of the discussers.

The author agrees that the 4-in.-wide slot in the test coupon, Fig. 37, is a major notch. The effect of variations in the depth of this notch has been evaluated by the load-eccentricity tests summarized in Table 4 of the paper, and the effect of variations in its width has been evaluated by the data given in Table 24 of this closure.

Roop inquires regarding the values of S_N and of the numerator of Equation [13], for notches of different angularity. It is obvious from the text of the paper that the value of S_N is 48,000 psi for the standard notch used. For other notches the value of the numerator is directly proportional to the angularity term given in Equation [13]. This is so apparent that its inclusion in Table 7 was felt to be unnecessary. In continuation, Roop also inquires regarding the notch angle at which the fracture changes to shear. This change occurs as the angle of notch exceeds about 90 deg, as is likewise stated in the paper. The computed value of the

factor for the angularity term for the 90-deg notch is 1.17. Accordingly, the nominal breaking strength S_N , of the steel tested containing such a notch would be $48,000 \times 1.17$, or about 55,150 psi. This is only slightly below the strength of the steel determined by the conventional tensile test, in which shear predominates. It is apparent that with the further increases in the angle of notch, shear fractures will be favored, as a result of which the breaking loads will increase due to the increasing amount of the cold work permitted by the enlarged notches. However, these data on nominal breaking strengths cannot be used to compute actual unit stresses without knowing the stress-concentration and other necessary factors involved in sections containing notches.

The temperature of test is stated in Table 8, because it differs from the standard test conditions given in the first part of the paper, under the section, "Determination of Effect of Variables on Development of Cleavage Fractures." Continuing his comments on the data given in Table 8, Roop discusses the possibility of evaluating the stress at which transition occurs from the cleavage to the shear type of fracture. As in the case discussed in the preceding paragraph, actual stresses at the notch cannot be evaluated without taking into consideration several important details peculiar to notched sections. Equation [14] indicates that the transition from cleavage to shear is not a discontinuous function. The same indication is shown by Fig. 41 of this closure, with respect to the shift in the transition temperatures by the change of the notch root radii. The notch-radius term, derived from the data of Table 8, is expressed as $2.085R^{0.113}$ in order to simplify Equation [15], since the derivation of this term is evident from the text of the paper.

In his subsequent comment, Roop discusses transition-temperature determinations by the author's tests and states that Kahn's tests yield the same results. This statement appears to be incorrect,³⁷ as is evident by the discussion of this paper by Kahn and Imbembo. In connection with the comments on the transition-temperature range, Roop's attention is called to the mathematical expressions covering the transition characteristics of the original four steels included in Fig. 15 of the paper, also of additional five steels included in Fig. 38, given in Equations [27] to [35] of this closure.

Several additional items are listed at the end of Roop's discussion. The first states that no effect of thickness is discernible in the tests given in Table 5, whereas in tests of other type thickness is a factor. The agreement of the P/A values with those of the theoretical equation, Equation [5], simply indicates that the geometry of the coupon and of the notch used minimizes the effect of the thickness. Equations [27] to [35] indicate some effect of the thickness in the transition characteristics of steels.

His next comment is to the effect that plastic deformation can precede cleavage fracturing. The author agrees that this is possible, especially if notches of large root radii are used. The statement made in the paper was intended to emphasize the fact that in cases where the fractures are "strictly" of cleavage type, such as those occurring at or below the temperature T_n , Fig. 49, no appreciable plastic deformation would be exhibited by the steel, with the entire test piece considered as a unit. The load-time charts in Fig. 50 illustrate the situation clearly. Some slight flow may occur at the surface, at localized areas or at locations where the stresses are biaxial, but on a macroscale no appreciable flow would be detected.

The third item states that very little scatter of results is shown in Figs. 8 to 11. In constructing these curves, the averages and

³⁷ Discussion by A. B. Bagsar of paper "A Method of Evaluating Transition From Shear to Cleavage Failure in Ship Plate, and Its Correlation With Large-Scale Plate Tests," by N. A. Kahn and E. A. Imbembo, *The Welding Journal*, vol. 27, 1948, Research Supplement, pp. 183-s-184-s.

not the individual test results were plotted; the respective tables show the variations in individual test results. Fig. 44 shows the degree of scatter obtained in steel 18 through the transition-temperature range, by plotting individual test results against testing temperature. It is the author's experience that with correct test procedure the P/A values yield more consistent results than measurements of load-extension or of energy of fracture propagation. Another factor which may account for less scatter of results consists in the fact that the geometry of the test coupon and of the notch used was based upon experimental data and were not chosen arbitrarily or on the basis of expediency.

Roop's fourth comment deals with the effect on the test results of the cold work present in the notch used. This subject has already been discussed in dealing with Jonassen's comments presented in the earlier section of this closure. In Table 21 and Fig. 46 are presented quantitative test data, showing that the effect of cold work at the notch is negligible in these tests.

His fifth comment is on the arrest of cleavage cracks by an interposing shear fracture, which has likewise been discussed in the treatment of Kinzel's comments relating to the same subject. The data given in the paper show that a definite load level is required to initiate cleavage-fracturing and that no fracture occurs if the loads are below this level. It is true that some ships have been reported to have fractured at very low stresses. Such reports may be in error because of the uncertainty of estimating the actual stresses under which fractures have developed; it is not necessary to assume that steels have an inherent property of fracturing in an unpredictable manner, under very low stresses.

W. Samans discusses the possible applications of the test data presented in the paper to pressure vessels and other fields. There is no reason to assume that the same mechanism of fracturing does not govern the performance of steel, regardless of the shape or service of the structure in which the steel is used. The particular failure described by Samans, which originated from a defect in the butt weld of a pressure vessel, conforms essentially to the same general rules given in the paper for cleavage fractures, although in the case cited the fracture originated from an internal rather than an external notch. Application of the elastic theory to the evaluation of the stresses in the types of notch used appears speculative.

The author agrees with J. W. Stewart that good technique and correct sequence of welding are essential for minimizing defects in welded ships. As is indicated by the data given in the paper, large cleavage fractures can originate from small defects or discontinuities. The use of riveted seams or straps as crack arresters, discussed by Stewart, has certain distinct disadvantages. These have been treated at length by the author in connection with McCutcheon's discussion of the same subject.

J. Gibson and C. F. Tipper compare the test results given in the paper with those obtained in the Engineering Department, Cambridge University, England, and state that some of the apparent differences are due to the use of different testing procedure and test coupon, with which statement the author agrees. In making this comparison they substitute in Equation [10] of the paper the term $a + (d + h)/2$ for load eccentricity e , and construct Fig. 34. The author's experience with the type of test coupon shown in Fig. 36, which is similar to that illustrated by Fig. 33, and used by Gibson and Tipper, has been summarized in the discussion of Bennett's comments in an earlier section of this closure. It should be observed that for test coupons 6 in. or less in depth, which is the case discussed by Gibson and Tipper, Equation [10] of the paper can be reduced to the form of Equation [5]

$$\frac{P}{A} = \frac{S}{1 + 6n \left(\frac{e}{d - h} \right)}$$

This equation is adequate for expressing the relationships of eccentricity of loading e , section depth $d - h$, unit breaking load P/A , and nominal breaking strength S , for the case under consideration. Both e and $d - h$ are variables, and P/A is a function of these two variables. For the sake of simplicity, the ratio of these two variables $e/(d - h)$ is used in the equation instead of the eccentricity component a (see Figs. 1 and 31). Thereby, the introduction of expressions involving positive and negative values is obviated. Aside from this, the use of e instead of a is preferred because of the direct relation of e to the centroidal axis of the test coupon. The eccentricity component a has been recognized and discussed in the paper, in connection with Equations [6] and [10].

It is apparent from the foregoing simplified equation that if the ratio $e/(d - h)$ equals 0.5, P/A is independent of the section depth and the loading is tangential to the vertex of the notch, or $a + h = 0$. As $e/(d - h)$ decreases from this value of 0.5, P/A increases. The latter corresponds to the case in which the term $a + h$ has a negative value described by Gibson and Tipper. For cases $e/(d - h) > 0.5$, P/A values decrease, which situation corresponds to positive values of $a + h$. In the tests reported in Table 4 of the paper the eccentricities and coupon depths used cover the cases $e/(d - h) > 0$ for coupons with a depth of 6 in., whereas those listed in Table 5 cover only cases in which $e/(d - h) > 0.5$. The P/A values obtained by the use of different $e/(d - h)$ ratios in all these tests conform to Equation [10], and the results obtained by Gibson and Tipper by their analysis likewise conform to the latter equation. If the ratio $e/(d - h)$ were plotted against the reciprocal of P/A , a straight line would be obtained, as is shown in Fig. 6 of the paper.

The next item discussed is the correction factor n in Equation [3]. The values of n were computed from A/P versus e curves shown in Fig. 6 of the paper, and are not based upon assumptions. It is obvious that n is the ratio of the relative stress-concentration factors for the axial tension and flexural stresses, as has been explained by Equations [36] and [37] given in the earlier section of this closure. The use of this ratio was preferred, since it appears to remain constant for different notch and coupon geometries, whereas the stress-concentration factors themselves do not.

The third item discussed by Gibson and Tipper deals with fracture loads. Although with the use of Izod notches ($R = 0.01$ in.) or of notches with large root radii, the fracture loads for shear and cleavage appear to be about the same, the difference between the loads for these two types of fracture is very pronounced if notches of small root radii, say of the order of 0.0015 in., are used. This is clearly shown by the data presented in Table 8 and Fig. 11, also by Equation [14] of the paper. This point is also explained by Figs. 38 to 43, and 49 and 50 of this closure, and by the respective tabulated test data. Recent work of numerous investigators,³⁸ using tensile coupons containing notches with root radii of the order of 0.001 in., has shown appreciably lower fracturing loads for cleavage. The charts in Fig. 50 also show that the pattern of the initiation of cleavage fractures is quite different from that of shear fractures. With the use of notches with small root radii, the overlapping type of fractures described by Gibson and Tipper may occur only within or near the transition range.

³⁸ "Distribution of Strength and Ductility in Welded Steel Plate as Revealed by the Static Notch Bar Tensile Test," by W. F. Brown, Jr., L. J. Ebert, and G. Sacks, *The Welding Journal*, vol. 26, 1947, Research Supplement, pp. 545-s-554-s.

Definitions of Heats of Combustion of a Fuel and Current Methods for Their Determination

By E. F. FIOCK,¹ R. S. JESSUP,¹ AND F. W. RUEGG¹

Definitions are given for high and low heats of combustion, both at constant pressure and at constant volume; and the relations connecting the four quantities thus defined are discussed. Apparatus in current use for measuring heats of combustion of solid, liquid, and gaseous fuels, the corrections which must be applied, and the procedure for calculating the results are described. A brief resume is given of the present prospect for calculating the heats of combustion of mixed gases from analytical data.

INTRODUCTION

THE usefulness of a fuel is determined by many complicated and interrelated factors, both economic and physical in nature. As examples, it seems unlikely that oil will entirely replace coal in regions where coal is plentiful or cheap, and no matter how attractive the compounds of boron may appear, such substances can never form the basis of our principal fuel supply because there simply is not enough boron in the earth's crust to fill the need. On the other hand, there is an extremely abundant supply of aluminum, so that the use of this metal as a fuel is not impossible. Suffice it to say here that availability is of paramount importance in determining what the fuel shall be for present and future power plants.

Normally a given type of power plant is designed to use a fuel of a more or less specific type. This is not meant to imply that there cannot be considerable variation in many of the physical properties within a single type for a specific operation. It does seem desirable, however, to question the rather widespread impression that a jet engine or gas turbine can use as a fuel any substance which will burn in air. Actually such power plants operate best only on the specific fuel for which they are designed. On the other hand, it seems probable that power plants of this type can be designed for satisfactory operation on a wide variety of fuels, much wider for example than is the case with reciprocating engines having either spark or compression ignition.

In addition to cost, availability, and engine design, the physical and chemical properties of the fuel are of great importance. In general, the nature of the application determines which properties are of paramount interest. In powered missiles and in aircraft, where fuel-tank volume is an important factor in determining range and maximum speed, it is highly advantageous and in some cases necessary to use a fuel which liberates the maximum possible heat per unit of its volume. On the other hand, there can be no great interest in the volume of tankage required for a stationary power plant.

Many familiar properties such as density, viscosity, vapor pres-

sure, and their rates of change with temperature, vary in importance with the application. Since the primary function of any power plant is the conversion of the latent chemical energy in the fuel into usable work, the burning characteristics of every fuel are basic, regardless of the proposed application. These characteristics include ignitability, ignition lag, rate of burning or flame speed, tendency to preignite and to detonate, expansion ratio, and heat of combustion. Of these, the last is probably the most useful single criterion of the prospective value of a fuel.

It is the purpose of this paper to discuss the heats of combustion of fuels, not from the standpoint of actual numerical values but by reviewing the common methods of measurement, the corrections involved, the significance of results obtained by each, and the interrelations of the results obtained by different methods.

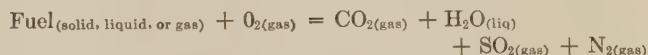
DEFINITIONS AND INTERRELATIONS OF HEATS OF COMBUSTION

For the expression "heat of combustion" various authors have used several synonymous terms including "calorific value," "heat or heating value," and "thermal value," each of these expressions being included by chemists in the more general term "heat of reaction."

In the ASTM Book of Standards, 1946 (1),² the expression calorific value (heat of combustion) and thermal value are used in connection with solid and liquid fuels (1a, b, and c), while calorific value alone is used in the section on gaseous fuels (1d). The present authors have personal preference for the expression "heat of combustion," and will adhere to it throughout this paper.

Broadly speaking, the heat of combustion means the quantity of heat liberated when unit quantity of fuel is burned. For any given fuel, the amount of heat which can be obtained by burning unit weight depends upon the initial state of the reactants and the final state of the products, so that a numerical value of heat of combustion will be definite only when the conditions to which it applies are specified.

The definitions of the various heats of combustion which have been found useful are all simply related to that of the quantity almost universally used in tabulations of heats of combustion of pure compounds. This quantity, for a fuel, is simply the heat evolved at constant pressure (or decrease in enthalpy, $-\Delta H_c$) in the following reaction:



This equation applies to the complete combustion in air or oxygen of unit quantity of a fuel containing any or all of the elements, carbon, hydrogen, oxygen, nitrogen, and sulphur, in addition to water and any constituents which may form ash. For the present purpose it is not considered desirable to complicate the definition of heat of combustion to the extent which is necessary to include certain special fuels and oxidants such as are used in some rocket motors.

The heat of the foregoing reaction depends somewhat upon the

² Numbers in parentheses refer to the Bibliography at the end of the paper.

¹ National Bureau of Standards, Washington, D. C.

Contributed by the Power Test Codes Committee as part of a Symposium on Heating Values of Fuels for Thermal Efficiency and Power Cycle Analyses, and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

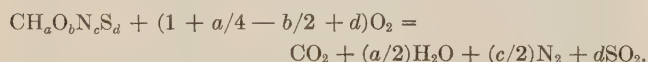
NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-123.

temperature of the reactants and products, and to a very slight extent upon their pressure. Hence it is desirable to agree upon standard or reference values of these parameters. Chemists generally use zero pressure for reference and others use 1 atm. The difference is negligible for most practical purposes. As to the reference temperature, chemists use 25 C (77 F), mechanical engineers 68 F, and in the gas industry 60 F is most common. Undoubtedly, each of these values was selected initially as an average value of room temperature, since there are many advantages in conducting calorimetric experiments at or near the temperature prevailing in the laboratory. The variation of heat of combustion with temperature over the range encountered in the laboratory is not great, and correction from the actual temperature to any of the afore-mentioned reference values can be made with ample accuracy.

The direct result of an experimental determination of heat of combustion is the heat produced by burning unit weight of fuel as measured, i.e., per unit weight of fuel plus ash plus any water which it may contain. The results of determinations on samples which contain significant amounts of water are usually reported on a dry basis (1a). To simplify the following discussion of the relations among the various heats of combustion, it is assumed that the measured values have been reduced to a water-free, ash-free basis. This discussion applies primarily to solid and liquid fuels, since the methods of relating the high and low heats of combustion of gaseous fuels are well standardized (1d, 16, 17).

For fuels which contain water or hydrogen, the final state of the water present after combustion is of considerable importance numerically. If all of the water present in the products of combustion is condensed to the liquid state, the total quantity of heat evolved is commonly designated as the "high," "gross," or "total" heat of combustion. If all this water remains in the gas phase, the heat liberated is called the "low" or "net" heat of combustion. In this paper the adjectives high and low will be used exclusively.

If the composition of the fuel is known quantitatively, it may be represented by the empirical formula $\text{CH}_a\text{O}_b\text{N}_c\text{S}_d$, in which the subscripts are not necessarily integers. Disregarding temporarily the states of reactants and products, the equation for the combustion of this fuel with oxygen is



If we consider the possible states of reactants and products we find the following: (a) The fuel may be either solid, liquid, or gaseous; (b) both the O_2 and the CO_2 will be gaseous; (c) the H_2O may be either liquid or gaseous, or distributed between these two phases; and (d) the N_2 and SO_2 will usually be gaseous, but in the oxygen-bomb calorimeter, the nitrogen may end up as nitric acid and the sulphur as sulphuric acid. Since the latter condition prevails in the calorimeter but not generally in the application, it is logical to correct the calorimetric values back to gaseous N_2 and SO_2 , as described later.

If combustion takes place at constant volume, and if the gaseous reactants are saturated with water vapor at room temperature and the products are cooled to this same temperature, it will be apparent that all of the water formed during combustion must condense to liquid. In gas calorimetry, the water formed in combustion is seldom all condensed, but the results are ordinarily corrected to the condition that it is all condensed. However, in applications of the combustion process it is rare indeed that condensation is completed in such a way that the heat liberated by this process can be utilized. Thus it is logical that many who wish to use numerical values of heat of combustion in calculating actual processes should choose the low value. The present authors feel that the choice between the high and the low value is simply a matter of habit and convenience, since the same end result will be

obtained from either, when proper account is taken of the initial and final states of the reactants and products.

Since there are reliable calorimeters which operate at constant volume and others which operate at constant pressure, and since there are practical applications which approximate each of these cases, the relation between the heat of combustion at constant volume and that at constant pressure must be considered. It is obvious that in a reaction at constant volume, no external work can be done by or on the enclosed system. However, for a reaction at constant pressure, external work will be done if there is a change in the volume of the total gas present, and in a calorimeter, the thermal equivalent of this external work will appear as a part of the observed heat of combustion. Therefore our definitions must state whether the process of burning has taken place at constant pressure or at constant volume.

Summarizing the requirements of a satisfactory definition of heat of combustion, we find that it must state the qualitative composition of the fuel and the products, the initial and final states, whether the water formed during combustion or present initially in the fuel is condensed to liquid or remains in the vapor state, and whether the reaction has been conducted at constant pressure or at constant volume. With these points in mind the following definitions are suggested:

1 The "high heat of combustion at constant volume" of a fuel containing any or all of the elements carbon, hydrogen, oxygen, nitrogen, and sulphur, is the amount of heat produced by the combustion of unit quantity of fuel at a standard temperature in an enclosure of constant volume, when the products of combustion, consisting of liquid H_2O and gaseous CO_2 , SO_2 , and N_2 , are cooled to the initial temperature.

2 The "low heat of combustion at constant volume," same definition as 1, except that all of the water formed during the combustion remains in the gas phase.

3 The "high heat of combustion at constant pressure" of a fuel containing any or all of the elements carbon, hydrogen, oxygen, nitrogen, and sulphur, is the amount of heat produced by the combustion of unit quantity of fuel at a standard temperature and at a constant pressure of 1 atm, when the products of combustion, consisting of liquid H_2O and gaseous CO_2 , SO_2 , and N_2 , are cooled to the initial temperature.

4 The "low heat of combustion at constant pressure," same definition as 3, except that all of the water formed during the combustion remains in the gas phase.

As to the quantity of fuel, it is common practice to report heats of combustion of solids or liquids on a weight basis, i.e., in Btu per pound. Gaseous fuels are reported on a volume basis, i.e., in Btu per standard cubic foot, which by general agreement means a cubic foot of fuel gas saturated with water vapor at 60 F, and at a total pressure of 30 in. of mercury absolute.

Although there is no permanent need for individual symbols for each of these heats of combustion, such symbols are needed here in the development of the relationships which exist among them. Table 1 which follows is not intended as a recommendation that this notation be used generally.

TABLE 1 HEATS OF COMBUSTION OF $\text{CH}_a\text{O}_b\text{N}_c\text{S}_d$

	Symbol	Usual method of measurement
High, at constant volume.....	$(h_h)_v$	Oxygen-bomb calorimeter
Low, at constant volume.....	$(h_l)_v$	
High, at constant pressure.....	$(h_h)_p$	Water-flow calorimeter
Low, at constant pressure.....	$(h_l)_p$	

If, for a reaction at the constant pressure p and the constant temperature T , the volume of gaseous products minus the volume of gaseous reactants is Δv and the volumes of solids and liquids are

neglected, external work $p\Delta v$ will be done by the system. Among other things, Δv will depend upon the state of the fuel and of the water. If neither of these is gaseous, their volumes may be neglected. It will be seen from the equation that, under these conditions, the change in the number of moles of gas accompanying the combustion of one formula weight (M) of fuel is $-a/4 + b/2 + c/2$. Inspection shows that sulphur does not enter the picture and that there will be a decrease in the number of moles of gas for all fuels except those high in oxygen and/or nitrogen. Thus for all ordinary fuels, work is done on the system by the atmosphere during combustion at constant pressure, and the high heat of combustion at constant pressure is greater than that at constant volume.

Assuming that the volumes of water and fuel are negligible and that the gases are perfect, the external work done by the system per formula weight of fuel is

$$p\Delta v = (-a/4 + b/2 + c/2)RT$$

Therefore for solid and liquid fuels we may write

$$(h_h)_v = (h_h)_p + (-a/4 + b/2 + c/2)RT/M$$

The difference between the high and low heats of combustion at constant volume is simply the internal energy of vaporization of the water formed per unit weight of the fuel. This amounts to 997 Btu per lb of H_2O (68 F). Thus

$$(h_1)_v = (h_h)_v - 997a/M$$

Similarly the difference between the high and low heats of combustion at constant pressure is the latent heat of vaporization of water formed per unit weight of fuel. This amounts to 1055 Btu per lb (68 F). Thus

$$(h_1)_p = (h_h)_p - 9503a/M$$

These relations can be transformed into others more convenient to use by introducing the weight fractions X , of the elements in the fuel, and the numerical values $R = 1.986$ Btu/mole deg F, and $T = 528$ deg R (68 F).

For the dry ash-free fuel it will be seen that $X_H = 1.008 a/M$; $X_O = 16b/M$; and $X_N = 14.008c/M$, from which $a = 0.992 MX_H$; $b = 0.0625MX_O$; and $c = 0.0714MX_N$.

Substitution of these values in the foregoing relations among the various heats of combustion gives

$$(h_h)_v = (h_h)_p - 260X_H + 33X_O + 37X_N \dots [1]$$

$$(h_1)_v = (h_h)_v - 8910X_H \dots [2]$$

$$(h_1)_p = (h_h)_p - 9430X_H \dots [3]$$

From these three equations, relating the four different heats of combustion, all may be calculated if any one is measured, and if the weight fractions of hydrogen, oxygen, and nitrogen in the original fuel are known.

As indicated previously, it has been assumed that the measured heat of combustion has been reduced to an ash-free dry basis. In practical applications, the quantity of most importance will be the heat of combustion of the fuel containing the same amounts of ash and water as at the time the fuel is used or marketed. The relations hold for a fuel containing ash when the heats of combustion are on a dry basis and when X_H , X_O , and X_N are the weights of hydrogen, oxygen, and nitrogen per unit weight of dry fuel plus ash. The heats of combustion of a fuel containing the fraction X_W of water will be given by the expressions $(h_h)_v(1 - X_W)$; $(h_1)_v(1 - X_W) - 997X_W$; $(h_h)_p(1 - X_W)$; and $(h_1)_p(1 - X_W) - 1055X_W$. In each of these expressions the first term represents the heat of combustion per unit weight of dry fuel plus ash, and the

complete expression represents the corresponding heat of combustion of fuel plus ash plus water.

The common methods for determining heats of combustion will be discussed next.

DETERMINING HEATS OF COMBUSTION OF SOLID AND LIQUID FUELS

Oxygen-Bomb Calorimeter. The heats of combustion of solid and liquid fuels are usually measured in an oxygen-bomb calorimeter. This instrument consists of a metal calorimeter vessel filled with water, in which are immersed a constant-volume bomb, a stirrer, and a thermometer for measuring the water temperature. The temperature rise of the calorimeter, produced by the combustion of a weighed sample of fuel in the bomb, is a measure of the heat of combustion of the fuel.

If the calorimeter vessel differs in temperature from its surroundings, there will be a transfer of heat between the calorimeter and the surroundings and a correction therefor must be applied. This correction can be evaluated only if the temperature of the surroundings is controlled, for example, by enclosing the calorimeter with a jacket whose temperature can be regulated. Such an arrangement is necessary if the accuracy of the observed values of heat of combustion is to be better than a few per cent.

A diagram of a typical bomb calorimeter, similar in construction to one described by Dickinson (2), is shown in Fig. 1 which is largely self-explanatory. The jacket cover shown, consisting of two sheets of copper separated by an air space and bent down at the edges to dip into the jacket water, was described by White (3), while the cover of Dickinson's calorimeter contained circulating jacket water.

The temperature of the calorimeter is usually measured by a calorimetric mercurial thermometer or by a Beckmann thermometer, such instruments being adequate when the accuracy of the calorimetric results need not be better than a few tenths of 1 per cent. If higher accuracy is required, a resistance or a thermoelectric thermometer may be used. The instrument shown in Fig. 1 is a resistance thermometer.

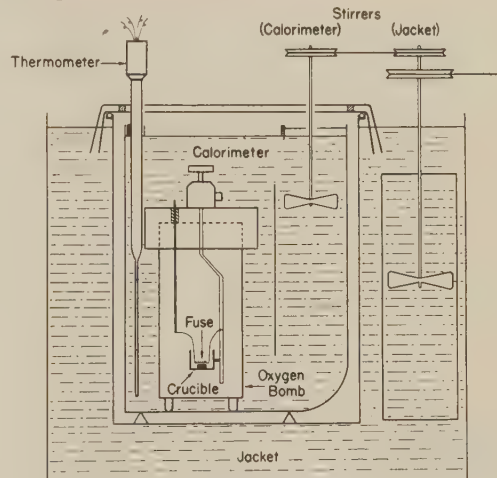


FIG. 1 SCHEMATIC DIAGRAM OF AN OXYGEN-BOMB CALORIMETER

Calorimetric resistance thermometers and bridges for use therewith have been described by Dickinson and Mueller (4), Sligh (5), Mueller (6), and Smith (7). Thermoelements and potentiometers suitable for calorimetric applications are described by White (8), and by Wenner (9). Most of the instruments mentioned in the references are now available commercially.

Oxygen to be used in a bomb calorimeter should be free of combustible impurities. Since commercial oxygen may contain enough of such impurities to cause an error of 1 per cent, it is necessary to remove them before placing the oxygen in the bomb. Usually this is done by passing the impure oxygen over hot copper oxide. One type of commercially available equipment for this purpose is described in reference (10).

Procedure. To measure heat of combustion with a bomb calorimeter, it is necessary to know the quantity of heat which produces a given temperature rise of the calorimeter. This may be determined by measuring the rise produced by burning in the calorimeter a known weight of a substance, such as benzoic acid, for which the heat of combustion (11) is known accurately. Measurements in which the temperature rise corresponding to a known amount of energy is determined are usually referred to as calibration experiments. The results of such experiments, expressed as the energy required to produce a rise of 1 deg in the calorimetric system, is called the energy equivalent or the effective heat capacity of the system.

This method of calibrating the calorimetric system eliminates systematic errors, such as might otherwise arise from the time lag in the distribution of heat throughout the system (12). It also makes the temperature scale unimportant, provided that same scale is used during both the calibration and the measurement of heat of combustion. It is important that calibration experiments and measurements of heats of combustion be made under the same conditions, as nearly as is practicable. In both types of experiment enough water (about 20 drops) is placed in the bottom of the bomb to insure initial saturation with water vapor. Since the temperature of the calorimeter rises only a few degrees, most of the water formed on combustion must condense to liquid, and the heat produced by the combustion of unit weight of sample is substantially equal to the high heat of combustion at constant volume. The small additional quantity of water which exists as vapor at the close of an experiment, due to the increased temperature of the bomb, is taken into account automatically in the calibration.

In preparing to make a run, a crucible containing a weighed sample of the unknown or of the standard fuel is placed in the bomb. Although coal can be burned in pulverized form, other solids including those used for standard substances, cannot, because part of the powder is frequently blown out of the crucible and fails to burn. Such substances can usually be burned completely if compressed into pellets.

Samples of liquids of low vapor pressure can be poured directly into the crucible in which they are to be burned, and weighed in the same manner as for solid fuels. For the more volatile substances, such as gasoline, it is necessary to enclose the sample in a suitable capsule to prevent loss by evaporation. Although gelatin capsules have been used, it is believed that thin-walled glass bulbs, flattened on opposite sides, filled completely, and sealed are more satisfactory (13). The flat sides are made flexible to prevent breakage of the capsule when oxygen is admitted to the bomb under pressure.

A fuse wire, coiled into a small helix, is attached to the electrodes of the bomb in such a way that the helix is nearly but not quite in contact with the sample. This wire may be iron or an appropriate alloy less subject to corrosion than iron. No. 34 wire is supplied by manufacturers of bomb calorimeters for this purpose. The bomb is then closed, flushed with oxygen free of combustibles, and charged with this gas to a pressure of from 20 to 40 atm, the mean of these values being used most generally.

The calorimeter vessel is placed on a scale, and water is added until the total weight reaches a previously selected value. During this process the temperature of the water is so adjusted that the final temperature of the calorimeter system after combustion

will be one or two tenths of a degree below the temperature of the jacket. The calorimeter is then placed within the jacket, the bomb with firing leads attached is positioned, the covers for calorimeter and jacket and the thermometer are put in place, and the calorimeter stirrer is started.

After a steady state is reached, the temperature of the calorimeter is observed at 1-min intervals for at least 5 min, to establish the rate of change of temperature due to stirring and heat transfer between calorimeter and environment. The sample is then fired by impressing approximately 10 volts a c or d c across the fuse wire, whereupon the latter burns. Solid or exposed liquid samples are ignited directly, and the combustion of the fuse wire is effective in breaking the glass bulbs containing liquid samples and then igniting the liquid contained therein.

Subsequent to ignition, a time record is kept of the calorimeter temperature at intervals as short as is practicable during the period of most rapid rise, and at 1-min intervals thereafter, until the rate of change has been constant for at least 5 min. The results obtained in this way are illustrated by the solid curve of temperature versus time shown in Fig. 2.

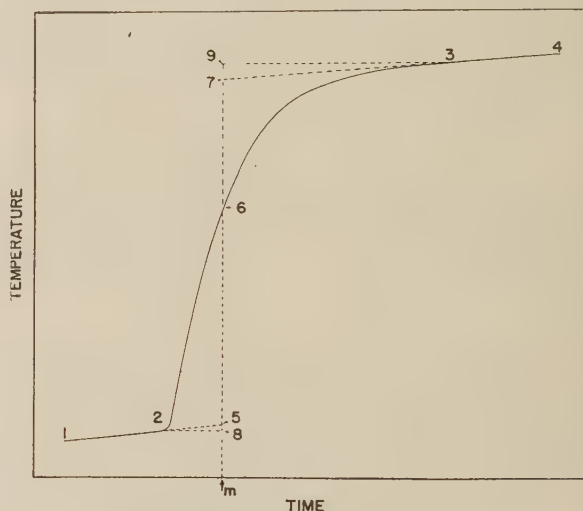


FIG. 2 TYPICAL TEMPERATURE-TIME CURVE FOR A BOMB-CALORIMETRIC EXPERIMENT

The bomb is then taken out, the pressure relieved, and the cover removed. All inner surfaces are rinsed thoroughly with distilled water, and the washings are titrated with 0.1 normal alkali solution, using methyl orange as the indicator, to determine the amount of acid formed during combustion. If the fuel is known to contain sulphur, or suspected thereof, the amount may be determined by the bomb-washing method described by the ASTM (14).

For most fuels, the amount of sulphuric acid formed on combustion can be estimated with sufficient accuracy as the difference between the total acid from the unknown fuel and the nitric acid from a fuel such as benzoic acid which contains no sulphur. A knowledge of the quantity of sulphuric acid formed is necessary in order to correct back to gaseous SO_2 , in accordance with our definition of heats of combustion.

If the adiabatic method is used, a correction for heat of stirring is required, and this can be determined from parts 1-2 and 3-4 of the curve in Fig. 2. If these parts have the same positive slope there is no significant exchange of heat with the surroundings, and the slope is due entirely to heat of stirring. A difference in the slopes indicates that adiabatic conditions are not strictly maintained, and that there is some heat transfer between calorimeter and environment.

Computation of Results. The procedure for calculating high heat of combustion, in accordance with the definition stated previously, from experimental results of the type described is as follows. Referring to Fig. 2, the observed rise in temperature produced by the burning is $T_3 - T_2$, where T_2 is the last temperature observed before ignition and T_3 is the first temperature observed after the rate of temperature change becomes constant, following the burning. The observed rise is due to the heat produced by the combustion of the fuel sample in the bomb, to the heat transferred between the calorimeter and its surroundings, and to the heat resulting from the stirring. A correction, δT , must be applied for the last two effects, in order to obtain the temperature rise due to combustion alone, and this can be calculated from the data obtained in the periods from t_1 to t_2 , and t_3 to t_4 . Changes in temperature during these periods are due solely to thermal leakage and to stirring, the latter of which may be assumed to add heat at a constant rate. It is also justifiable to assume that the rate of thermal leakage is directly proportional to the difference between the temperature of the calorimeter T , and the constant temperature of the surroundings T_0 .

If b is the constant rate of change of temperature due to stirring alone and a is the thermal-leakage modulus of the calorimeter, then

$$dT/dt = b - a(T - T_0) \dots \dots \dots [4]$$

and if T_k is the temperature which would be attained in infinite time if both T_0 and b remain constant, Equation [4] may be written

$$dT/dt = -a(T - T_k) \dots \dots \dots [5]$$

A value of the slope dT/dt and the corresponding mean value of the calorimeter temperature T can be obtained from the data illustrated by each of the branches 1-2 and 3-4 in Fig. 2. Substituting these values in Equation [5] gives simultaneous equations which may be solved for a and T_k . The temperature change due to thermal leakage and stirring during the interval between t_2 and t_3 will then be

$$\delta T = -a \int_{t_2}^{t_3} [T(t) - T_k] dt \dots \dots \dots [6]$$

in which $T(t)$ represents temperature as the observed function of time, along section 2-3 of the curve. The temperature rise due to combustion alone is then

$$\Delta T = T_3 - T_2 - \delta T \dots \dots \dots [7]$$

and the quantity of heat Q , produced by the combustion is

$$Q = C \Delta T \dots \dots \dots [8]$$

in which C is the energy equivalent of the calorimeter, as determined in a calibration experiment.

The correction δT can be evaluated by graphical integration of the curve in Fig. 2, but this practice is tedious, and a simpler method, such as that suggested by Dickinson (2), is used more often. In this method, horizontal lines are drawn from points 2 and 3 to intersect a vertical line t_m , which is so located that the areas 2-8-6 and 6-9-3 are equal. The temperatures T_6 and T_7 are then calculated from the relations

$$T_6 = T_2 + (T_2 - T_1)(t_m - t_2)/(t_2 - t_1)$$

and

$$T_7 = T_3 + (T_4 - T_3)(t_m - t_3)/(t_4 - t_3)$$

It can be shown that the difference $T_7 - T_6$, obtained in this way, is the corrected temperature rise ΔT .

If it were necessary to determine the areas 2-8-6 and 6-9-3 by

graphical integration, this method would have no advantage over the direct integration of Equation [6]. However, Dickinson showed that the time t_m corresponds to a temperature rise which is a nearly constant fraction of the total temperature rise $T_3 - T_2$, and that once this fraction is determined, a few observations of temperature and corresponding time taken during the period of rapid rise suffice to determine t_m by interpolation. The fraction of the total rise corresponding to t_m is known to vary slightly with such factors as the mass of the bomb and the rate of stirring, but no significant error is introduced by assuming that this fraction is 0.60, provided this same value is used in both the calibration experiments and in the measurements of heat of combustion.

When the method is made adiabatic by adding heat to the jacket so that its temperature is kept equal to that of the calorimeter at all times, and when there is no heat leakage through the thermometer, leads, etc., the only correction to the observed temperature rise is that due to stirring. This correction, to be subtracted from the observed rise, is simply the product of the time interval $t_3 - t_2$ and the constant slope of either line 1-2 or 3-4.

If the adiabatic method is used and the jacket does not completely prevent heat interchange between the calorimeter and the room, this fact will be indicated by a difference in the slopes of lines 1-2 and 3-4. In this case the correction for the combined effect of heat transfer and heat of stirring can be made by the method described for the ordinary nonadiabatic case.

Although most of the corrected temperature rise is due to the heat of combustion of the fuel sample, small contributions are also made by the electrical energy supplied to ignite the fuse wire, by the combustion of this wire, and by the formation of nitric and sulphuric acids during combustion in high-pressure oxygen. The total energy added to the calorimeter in igniting and burning the fuse is determined by blank runs in which nothing else is burned. This correction can be made as small as 5 cal (0.02 Btu) by using approximately 1 in. of fuse wire. The correction for nitric acid is 1.41 cal (0.0056 Btu) per ml of 0.1 normal alkali neutralized.

If the fuel contains sulphur, all of the acid formed is determined in terms of 0.1 normal alkali, and a correction is applied just as if all of the acid were nitric. An additional correction amounting to 1300 cal per gm (2340 Btu per lb) of sulphur, converted into sulphuric acid, is then applied to account for the excess of the energy of formation of sulphuric acid over that of an equivalent amount of nitric acid, and for the difference in the energy of formation of aqueous sulphuric acid and gaseous SO_2 , as specified in the previous definition.

The corrections for firing energy and for acid formation are subtracted from the Q already calculated from Equation [8]. This difference, divided by the known weight of the fuel sample, is the high heat of combustion at constant volume, in accordance with the definition already given for this property.

In a calibration experiment made with a fuel of known high heat of combustion at constant volume and free of sulphur, the observed temperature rise is first corrected as already described. Corrections for firing energy and for the energy of formation of nitric acid are added to the product of the weight of the standard sample and its high heat of combustion at constant volume. The energy equivalent of the calorimeter system C , is this corrected energy divided by the corrected temperature rise.

Some factors other than those which have been enumerated are known to have small effects upon heats of combustion determined by the oxygen-bomb method. Among these are the initial pressure of the oxygen in the bomb, the weight of fuel burned per unit volume of the bomb, and the quantity of water placed in the bomb before the sample is burned. These factors have been discussed by Washburn (15), and need not be treated in de-

tail here because their total effect seldom amounts to one tenth of one per cent of the measured heat of combustion.

DETERMINING HEATS OF COMBUSTION OF GASEOUS FUELS

Heats of combustion of gaseous fuels are usually measured either with a water-flow calorimeter of the Junkers type, or with a recording calorimeter.

Water-Flow Calorimeter. The essential features of a water-flow calorimeter are shown diagrammatically in Fig. 3 which also includes such auxiliaries as a regulator for controlling the gas pressure, and a wet meter for measuring its volume, as well as a balance for weighing the water which is heated. Other necessary equipment not shown includes a psychrometer, a barometer, apparatus for analyzing the fuel and the flue gases, a stop watch, and a vessel of known volume for calibrating the wet meter. The important parts of the calorimeter proper will be indicated as its operation is described.

Fuel gas is supplied at a measured, constant rate to a burner of the Bunsen type. The primary air enters through adjustable ports near the bottom of the burner tube, these ports being adjusted by visual observation of the flame while the burner tube is outside the calorimeter. Radiation shields are mounted on the burner tube, and both the tube and the shields have polished surfaces of low emissivity to reduce the loss of heat downward. The secondary air enters through a large port around the burner tube. Its upward progress is facilitated by staggered perforations in the radiation shields.

Beyond the flame which is anchored on the burner tip, the products of combustion rise vertically through the combustion chamber for about 1 ft, until they impinge upon a conical deflector dome which causes them to flow downward through a large number of parallel tubes of a heat interchanger. The exterior surfaces of the combustion chamber, deflector dome, and interchanger tubes are all bathed in flowing water. The cooled products of combustion from the interchanger tubes are collected in an annular space, from which the liquid water leaves through a drain tube, and the gaseous products escape to the atmosphere through a side tube having an opening for a thermometer or sampling tube, and a damper. This damper controls the size of the exit opening, and hence also the rate at which the secondary air can enter the calorimeter.

The water passages of the calorimeter are shaded in Fig. 3. The inlet head is made independent of the rate of water supply by using a weir, and the rate of flow through the instrument under this constant head is controlled by a valve. The cooling water leaves the instrument over a second weir, whence it is collected in a tank which is weighed to determine the quantity flowing during an integral number of revolutions of the pointer of the wet meter. The temperatures of the water entering and leaving the instrument are measured with thermometers located as shown. A series of mixing baffles is provided ahead of the outlet thermometer to promote uniformity of the temperature at this location.

A quantity which may be called the "observed heating value of the gas" is calculated as the product $W\Delta T/VF$, in which W is the number of pounds of water heated by burning V cubic feet of fuel gas saturated with water vapor and measured at the temperature and pressure prevailing in the wet meter; F is a factor to reduce V to saturation at 60 F and 30 in. of mercury; and ΔT is the rise in temperature of the water in degrees F.

A number of corrections must be applied to the observed heating value in order to obtain the heat of combustion. These corrections depend to some extent upon the conditions of operation of the calorimeter, the composition of the gas, the temperature and relative humidity of the air in the room, and to a small degree upon the barometric pressure. Application of these corrections

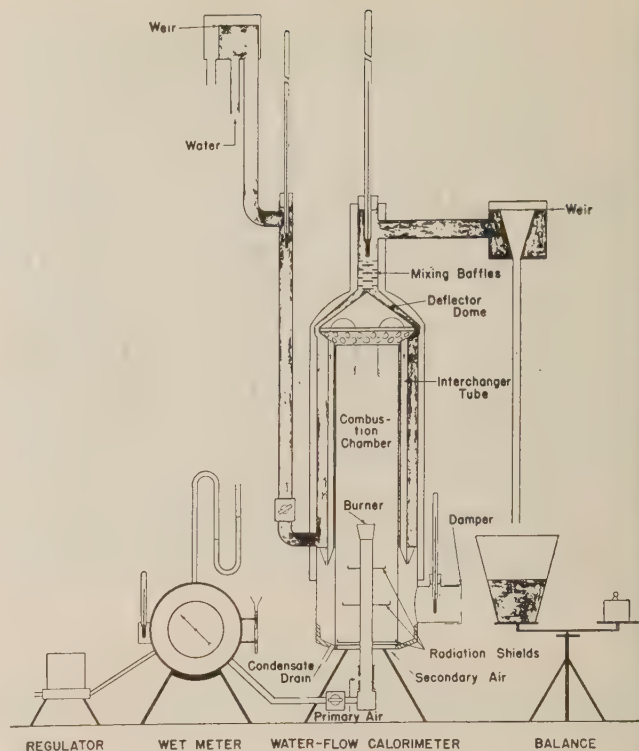


FIG. 3 SCHEMATIC DIAGRAM OF A WATER-FLOW CALORIMETER AND ACCESSORY EQUIPMENT

has been simplified by adopting the following standard conditions for the operation of the calorimeter:

- (a) Gas is burned at the thermal rate of 3000 Btu per hr (high heat of combustion).
- (b) The rate of water flow is adjusted so that the observed rise in its temperature is approximately 15 deg F.
- (c) The temperature of the inlet water and that of the wet meter are maintained as nearly equal to that of the room as is practicable.
- (d) The damper is adjusted so that the quantity of air entering the calorimeter exceeds the theoretical requirement for complete combustion by about 40 per cent.

The principal correction required is for the difference which may exist between the quantity of water vapor which enters with the air and gas and that which leaves the calorimeter through the damper, per unit volume of fuel. In general, more water vapor leaves than enters, and, under definite operating conditions such as those just specified, there is one value of the relative humidity of the entering air for which the humidity correction will be zero. Hence the necessity for making the correction can be eliminated by controlling the humidity of the inlet air properly. In the absence of such control, the humidity correction is readily calculable from the observed temperatures of air, gas, and flue gases, from the known ratios of the volumes of air and flue gases to the volume of gas burned, and from the observed relative humidity of the air, together with the known heat of vaporization of water. It usually amounts to less than 2 per cent, but may be as high as 5 per cent under conditions of high temperature and low relative humidity of the atmosphere.

When highest accuracy is desired, minor corrections must be applied for the buoyancy of the air upon the water which is weighed, the variation of the specific heat of water with temperature, and the loss of heat from the surface of the calorimeter.

The algebraic sum of these minor corrections seldom exceeds 0.4 per cent of the high heat of combustion. If the temperature of the entering water differs from that of the room, a further correction amounting on the average to about 0.2 per cent per deg F difference is necessary.

If the low heat of combustion of a gas is to be determined by this method, the same procedure is followed, but, in addition, the water condensed during the burning of a known volume of gas is collected at the condensate drain and its quantity determined. The heat of vaporization of the water thus collected is subtracted from the observed heat of combustion. No humidity correction is made in this case, but the other corrections mentioned are applied, and the result is the low heat of combustion of the fuel gas.

Detailed directions for determining the heats of combustion of fuel gases with the water-flow calorimeter, and for calculating the results of such determinations have been published by the ASTM (1*d*), by the Pacific Coast Gas Association (16), and by the National Bureau of Standards (17). Less complete directions are contained in the ASME Test Code for Gaseous Fuels (18).

Recording Gas Calorimeter. The only recording calorimeter in general use in this country is manufactured by Cutler-Hammer, Inc., Milwaukee, Wis.

In this instrument, gas is burned at a constant rate, and the heat produced is absorbed by a stream of air. The rates of flow of gas, air for combustion, and heat-absorbing air are regulated by metering devices similar in construction to ordinary wet-gas meters. These metering devices are geared together and driven by an electric motor, so that the relative proportions of the gases remain constant. The products of combustion are cooled nearly to the initial temperature of the gas and air in a heat exchanger, and do not mix with the heat-absorbing air. Since the water formed by combustion is condensed to liquid, the rise in temperature of the heat-absorbing air is proportional to the high heat of combustion of the fuel. This rise in temperature, as indicated by nickel resistance thermometers, is translated into Btu per standard cubic foot and recorded.

The manufacturer's book of instructions describes in detail the steps which are necessary to standardize the instrument by adjusting the recorder and the proportioning system, and by burning pure hydrogen for reference purposes. An investigation (19) of an early model of the recording calorimeter indicated that the results which could be obtained by its use agreed with those obtained with a water-flow calorimeter within the accuracy of either instrument.

Method of Chemical Analysis. In some instances it may be found desirable to calculate the heat of combustion of a gaseous fuel from its composition determined analytically, and the known heats of combustion of the individual constituents. The possibilities and limitations of this method were investigated recently in a survey sponsored by Subcommittee D-3-VII of the ASTM, in which identical samples of natural gas and of carbureted water gas were submitted for analysis to many participating laboratories. Some of these used conventional chemical methods and some the mass spectrometer.

For the present purpose, the published results on natural gas

(20) lead to so nearly the same conclusions as those on carbureted water gas (21) that it will suffice to summarize only the former (Table 2). These results indicate that reliable values of heats of combustion can be calculated from analytical data, but that the analyses must be conducted with considerable attention to detail, if errors which may otherwise be as high as 5 per cent are to be avoided.

BIBLIOGRAPHY

- 1 Book of Standards, part III-A, American Society for Testing Materials, 1946, (a) "Methods of Laboratory Sampling and Analysis of Coal and Coke," designation D 271-46; (b) "Method of Test for Thermal Value of Fuel Oil," designation D 240-39; (c) "Definition of the Terms Gross Calorific Value and Net Calorific Value of Fuels," designation D 407-44; (d) "Method of Test for Calorific Value of Gaseous Fuels by the Water-Flow Calorimeter" (tentative), designation D 900-46T.
- 2 "Combustion Bomb Calorimetry and the Heats of Combustion of Cane Sugar, Benzoic Acid, and Naphthalene," by H. C. Dickinson, Bulletin, National Bureau of Standards, vol. 11, 1914, pp. 189-257.
- 3 "The Modern Calorimeter," by W. P. White, Reinhold Publishing Corporation, New York, N. Y., 1928.
- 4 "New Calorimetric Resistance Thermometers," by H. C. Dickinson and E. F. Mueller, Bulletin, National Bureau of Standards, vol. 9, 1913, pp. 483-492.
- 5 "Recent Modifications in the Construction of Platinum Resistance Thermometers," by T. S. Sligh, Jr., Scientific Papers of the National Bureau of Standards, vol. 17, 1922, pp. 49-63 (Sci. Paper No. 407).
- 6 "Wheatstone Bridges and Some Accessory Apparatus for Resistance Thermometry," by E. F. Mueller, Bulletin, National Bureau of Standards, vol. 13, 1916, pp. 547-561.
- 7 "On Bridge Methods for Resistance Measurements of High Precision in Platinum Thermometry," by F. E. Smith, *Philosophical Magazine*, vol. 24, 1912, pp. 541-569.
- 8 "Thermoelement Installations, Especially for Calorimetry," by W. P. White, *Journal of the American Chemical Society*, vol. 36, 1914, pp. 1856-1868.
- 9 "Potentiometers for Thermoelectric Measurements Especially in Calorimetry," *ibid.*, pp. 1868-1885.
- 10 "Thermal Technic," by W. P. White, *Review of Scientific Instruments*, vol. 4, 1933, pp. 142-146.
- 11 "Note on Potentiometer Design," by F. Wenner, *Physical Review*, vol. 31, 1910, p. 94.
- 12 "A Method of Reducing the Effect of Disturbances in the Galvanometer Branch of a Potentiometer Circuit," by F. Wenner, National Bureau of Standards, *Journal of Research*, RP 1194, vol. 22, 1939, pp. 425-430.
- 13 "The Wenner Potentiometer," by L. Behr, *Review of Scientific Instruments*, vol. 3, 1932, pp. 109-120.
- 14 "A Potentiometer for Measuring Voltages of 10 Microvolts to an Accuracy of 0.01 Microvolt," by R. P. Teele and S. Schuhmann, National Bureau of Standards, *Journal of Research*, RP 1195, vol. 22, 1939, pp. 431-439.
- 15 "Heats of Isomerization of the Five Hexanes," by E. J. R. Prosen and F. D. Rossini, National Bureau of Standards, *Journal of Research*, RP 1420, vol. 27, 1941, pp. 289-310.
- 16 "Heat of Combustion of Benzoic Acid, With Special Reference to the Standardization of Bomb Calorimeters," by R. S. Jessup, National Bureau of Standards, *Journal of Research*, RP 1499, vol. 29, 1942, pp. 247-270, and the references cited therein.
- 17 "Note on the Density and Heat of Combustion of Benzoic Acid," *ibid.*, RP 1711, vol. 36, 1946, pp. 421-423.
- 18 "Note on Temperature Correction Methods in Calorimetry," by R. S. Jessup, *Journal of Applied Physics*, vol. 13, 1942, pp. 128-137; also see reference (3), p. 88.
- 19 "Heats of Combustion of the Liquid Normal Paraffin Hydrocarbons From Hexane to Dodecane," by R. S. Jessup, National Bureau of Standards, *Journal of Research*, RP 966, vol. 18, 1937, p. 115-128.
- 20 "The Heats of Combustion of Aromatic Hydrocarbons and Hexamethylene," by T. W. Richards and F. Barry, *Journal of the American Chemical Society*, vol. 37, 1915; pp. 993-1020.
- 21 See Reference (1a), p. 35.
- 22 "Standard States for Bomb Calorimetry," by E. W. Washburn, National Bureau of Standards, *Journal of Research* RP 546, vol. 10, 1933, pp. 525-558.
- 23 "Fuel Gas Calorimetry—Water-Flow Method," Supplement to vol. 32, Proceedings of the Pacific Coast Gas Association, 1941.

TABLE 2 RESULTS OF ANALYSES OF NATURAL GAS

	Specific gravity at 60 F, 30 in. Hg	High heat of combustion, Btu per cu ft
Direction determination.....	0.6820 ± 0.00005	1103 ± 4
Results of chemical analysis:		
Range of values.....	0.630 to 0.710	1057 to 1153
Average value.....	0.672	1101
Per cent of values averaged.....	88	85
Results of spectrometer analysis:		
Range of values.....	0.673 to 0.701	1088 to 1130
Average value.....	0.681 ± 0.004	1111
Per cent of values averaged.....	97.5	100

This publication is a report of the Calorimetry Standards Committee of the PCGA.

17 "Industrial Gas Calorimetry," by C. W. Waidner and E. F. Mueller, National Bureau of Standards, Technologic Paper No. 36, 1914.

"Standard Methods of Gas Testing," Circular No. C48, National Bureau of Standards, 1916.

"Gas Calorimeter Tables," Circular No. C417, National Bureau of Standards, 1938 (Revised edition soon to appear as Circular C464).

18 "Test Code for Gaseous Fuels," ASME Power Test Codes, New York, N. Y., 1944.

19 "The Thomas Recording Gas Calorimeter," by R. S. Jessup, National Bureau of Standards, *Journal of Research*, RP 519, vol. 10, 1933, pp. 99-121.

20 "Analysis of a Natural Gas by Volumetric Chemical Methods and by Mass Spectrometer," by M. Shepherd, *Analytical Chemistry*, vol. 19, 1947, pp. 635-640.

21 "Analysis of a Standard Sample of the Carburetted Water-Gas Type by Laboratories Co-Operating With the American Society for Testing Materials," by M. Shepherd, National Bureau of Standards, *Journal of Research*, RP 1704, vol. 36, 1946, pp. 313-349.

Uses of High- and Low-Heat Values

By A. G. CHRISTIE,¹ BALTIMORE, MD.

This paper gives current definitions of thermal efficiency and current usage of observed fuel heat values, and consumption test data for the evaluation of thermal efficiencies reported for (a) steam and (b) internal-combustion power plants. The author summarizes uses of high- and low-heat values in American practice. Commercial practice as well as theoretical considerations have influenced the choice of high- or low-heat values for the determination of thermal performances.

THERMAL EFFICIENCY

THE thermal efficiency of power-generating equipment involving fuels is the ratio of the output expressed in equivalent Btu to the heat value of the fuel supplied. In British and Continental practice, this ratio is expressed as a per cent. In American and Canadian power circles, power-plant performance is expressed as "Btu per kilowatt-hour," which is found by multiplying the number of units of fuel furnished per kilowatt-hour by the heat value of that fuel unit. Our present interest is the consideration of the heat value of fuel that is to be used in expressing thermal performance.

HEAT VALUES IN AMERICAN PRACTICE

Engineers are interested in the thermal performance of power-generating equipment either for the verification of purchase guarantees or as a check on daily plant operation. Tests to determine such performances are, in general, made in conformity with the ASME Power Test Codes. Therefore these codes can be considered as representative of the best engineering thought in America on the question of proper heat values for different fuels to express such performances. Let us examine the requirements of the several codes.

The Code on Definitions and Values, 1945, Par. 134, "Heat Value," discusses the differences between heat values as determined by calorimetric measurements at constant pressure and at constant volume. The variations between these heat values are presented and the discussion is summarized as follows:

"Because of these small variations, the heat value of a fuel as determined by either a constant-pressure or constant-volume calorimeter at ordinary atmospheric temperatures is acceptable for engineering calculations regardless of the actual conditions of burning the fuel in service."

High- and low-heat values are defined in the Code on Definitions and Values, as follows:

"The high-heat value at constant pressure of a fuel may be defined as the heat transferred from the products of complete combustion per unit mass of a solid or liquid fuel, or per unit standard volume of a gas fuel, when the products are cooled to the initial temperature of the oxygen or air-fuel mixture and includes the enthalpy of vaporization of all the water of combustion from

the hydrogen in the fuel. The pressure and temperature may be taken at the prevailing atmospheric values.

"The low-heat value at constant pressure is defined the same as the high except that the enthalpy of vaporization of the water of combustion is excluded from the heat transferred on the assumption that the water of combustion is not condensed. (Reduction factor of 1030 Btu for 1 lb of H₂O at 68 F.)

"For gas fuels the unit standard volume is one cubic foot of gas at one standard atmosphere and standard temperature, the gas having a relative humidity of 100 per cent.

"In order to avoid confusion, the term 'heat value' must always be qualified by the words 'high' or 'low'. Heat input, thermal efficiency, etc., may be computed on the basis of low-heat value or high-heat value or both. In commercial practice, the high-heat value is generally employed for solid and liquid fuels; but in the case of gas, the efficiencies and heat ratio calculated from low-heat value do not vary appreciably for wide changes in the character of gas; the corresponding calculation with high-heat values do. The individual test codes direct which basis should be used.

"The low-heat value is also used for such calculations as for flame temperatures and other calculations for which the high-heat value does not apply."

These definitions form the basis for the standards of heat values in the several ASME Power Test Codes. Thus the Test Code for Solid Fuels, 1929, Par. 19, states:

"In calorific determinations, the high heating value shall be used for solid fuels. This shall be stated in terms of Btu per pound of fuel on 'as received' or 'as used' basis."

The Test Code for Liquid Fuels, 1921, Par. 6, reads as follows:

"Calorific value shall be stated as 'total' heating value."

Reference is made to ASTM Standard Method of Test for Thermal Value of Fuel Oil, D 240-27, which specifies a test by bomb calorimeter. Hence "total" heating value must be high-heat value.

In the Test Code for Gaseous Fuels, 1944, Par. 1, reads as follows:

"The American and Sargent calorimeters are most commonly used in this country and shall be employed for the determination of gas-fuel heat values, unless a continuous record as produced by the Thomas recording calorimeter is desired."

Either high- or low-heat values can be found by the American or Sargent calorimeters. The Thomas recording gas calorimeter gives high-heat values. The code also gives tables of both high- and low-heat values of gases which may be employed to find unit heat values if a gas analysis is available.

The Test Code for Stationary Steam-Generating Units, 1946, states in its "Description and Definitions of Terms," Par. 16:

"Heat value of fuel; H_f : the high-heat value of the fuel."

The Test Code for Internal-Combustion Engines is now under revision. The tentative draft of this code recognizes both the high- and low-heat values of fuels. A paragraph in the Code reads as follows:

"The high-heat value of fuels containing hydrogen includes some heat not available for conversion into work in any internal-combustion engine. The difference between the high- and low-

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Contributed by the Power Test Codes Committee as part of a Symposium on Heating Values of Fuels for Thermal Efficiency and Power Cycle Analyses, and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-123.

heat values of commercially available petroleum used in liquid-fuel engines is a fairly constant percentage.

"For liquid-fuel engines it is, therefore, customary to base heat consumption guarantees upon the high-heat value. For gaseous fuels, however, the percentage difference between the high- and low-heat values may range from zero to more than 15 per cent, because of varying hydrogen content. It is customary, therefore, to base guarantees of heat consumption performances for gas and dual-fuel engines upon the low-heat values. In order to avoid confusion, the term 'heat value' must always be qualified by the word 'high' or 'low.'"

The Test Code for Steam Locomotives, 1941, states in Par. 24 that calorific determinations on the fuel are to be made in accordance with ASME Test Code for Solid Fuels. As noted previously, this specifies the high-heat value.

A tentative draft of the Test Code for Gas-Turbine Power Plants states in the "Description and Definition of Terms:"

"Plant Heat Rate—Plant fuel rate multiplied by the higher heating value of the fuel."

These paragraphs summarize American practice in the use of heat values to determine thermal efficiencies as exemplified by the ASME Power Test Codes.

COMMERCIAL USES OF HEAT VALUES

Solid and liquid fuels are sold on the basis of high-heat values per unit. Gaseous fuels are also sold on high-heat values which in many utilities are continuously checked by Thomas recording calorimeters. These heat values are readily determined by bomb or flow calorimeters without the necessity of a chemical analysis to determine the hydrogen content.

Some engineers have held that high-heat values should be standard for thermal-efficiency determinations since fuels are purchased on the basis of high-heat values.

Another argument in favor of using the high-heat value is that there is no more reason for neglecting the losses due to the latent heat of water vapor in the exhaust resulting from burning hydrogen than there is reason to neglect the heat carried away by the condenser cooling water in figuring the over-all efficiency of a steam cycle.

On the other hand, designers of engines using gaseous fuels maintain that it is difficult to relate the performance of a gas engine with the guarantees when high-heat values are used, and the gas used on test differs in hydrogen content from the one

specified in the contract. It is held that thermal performances are comparable when measured in terms of low-heat values, since allowance is then made for the unavailable energy of the fuel, which is due to the inability of the engine to recover the latent heat in the water vapor formed from the burned hydrogen.

Roughly stated, the low-heat values for bituminous coals range from about 3 to 4 per cent below the high-heat values. In liquid fuels of petroleum base, the difference is about 6-7 per cent. With gaseous fuels the difference may vary from a small value to as high as 15 per cent in the case of nearly pure hydrogen.

On the continent of Europe it is general practice to express thermal efficiencies on the basis of low-heat values. Their efficiencies thus stated with the same fuel, therefore, will exceed those based on high-heat values by the percentages just noted. This must be kept in mind when considering guarantees or test performances of continental prime movers and allowances made for these percentage differences.

Engineers use low-heat values in computations of flame temperatures and combustion-chamber performances, and in calculations involving heat absorbers, as boilers. Its use for such design purposes is fully justified as the latent heat of the water vapor resulting from the combustion of the hydrogen is not available in these cases.

A new term has been introduced in gas-turbine work, viz., "heating value at combustor outlet temperature." This heat value has been defined by M. A. Mayers and W. W. Carter.² It is found by subtracting from the high-heat value of the fuel used at room temperature, the increases in enthalpy of the products of combustion from room to combustor temperatures minus the calculated increase in enthalpy between these temperatures of the oxygen consumed. This expression has been introduced as a measure of the effectiveness of the combustor which is an important element in gas-turbine performance.

CONCLUSION

The foregoing constitutes a brief summary of the uses of high- and low-heat values in American practice. Both have a legitimate place in engineering. Commercial practice as well as theoretical considerations have influenced the choice of high- or low-heat values for the determination of thermal performances.

² "The Elbow Combustion Chamber," by M. A. Mayers and W. W. Carter, Trans. ASME, vol. 68, 1946, pp. 391-398.

Energy-Temperature Relations in the Combustion of Fuels in Gas Turbines

By R. V. KLEINSCHMIDT,¹ STONEHAM, MASS.

In this paper, the author discusses procedures for evaluating energy, temperature, and pressure relations based on precise knowledge of thermodynamic properties of the gases resulting from the combustion of fuel in thermal devices, including gas turbines. The problem of computing thermal efficiency and fuel required to cause a given temperature rise in the working fluid of a gas turbine is discussed.

THE concept of heating value of a fuel as the "heat of reaction" at room temperature, or more accurately, the enthalpy of reaction (at constant pressure) or the internal energy of reaction (at constant volume) was satisfactory for most engineering purposes, when the chief use for fuels was for production of steam, with air and water entering the system at "room temperature." The lowest temperatures at which heat was utilized and rejected were room temperatures. There arose some question as to whether the power plant could be charged with latent heat of condensation of moisture which it could not use, especially since condensation of this moisture is deleterious to the heating surfaces and therefore is avoided.

With the advent of internal-combustion engines, further difficulties arose, especially in connection with Diesel engines, since it was known that, theoretically at least, engines operating on fuels of different chemical composition would have different thermal efficiencies if computed on the basis of either higher or lower heating value. However, the errors were not large in most cases and were frequently attributed to or masked by varying combustion conditions with the different fuels.

GAS-TURBINE CONSIDERATIONS

With the advent of gas turbines, however, the problem of computing thermal efficiency and fuel required to cause a given temperature rise in the working fluid becomes more serious. To see how this is brought about let us review briefly the physical chemistry of the combustion process.

In Fig. 1 we have a temperature-enthalpy diagram. In this diagram we include not only the ordinary thermodynamic energies, work and heat, but also the chemical energy of the system which is to be converted into heat in the combustion process. Then the combined enthalpy of the fuel and air entering the system at room temperature is represented by point (1). We then burn the fuel and air at constant pressure and remove the "heat of combustion" so that the products of combustion return to the initial temperature, but at a lower enthalpy, represented by point (2). The difference in enthalpy is then the higher heating value ($H_1 - H_2$) at constant pressure.

Now supposing that we have air and fuel already heated to a high temperature, by adiabatic compression or heat exchange

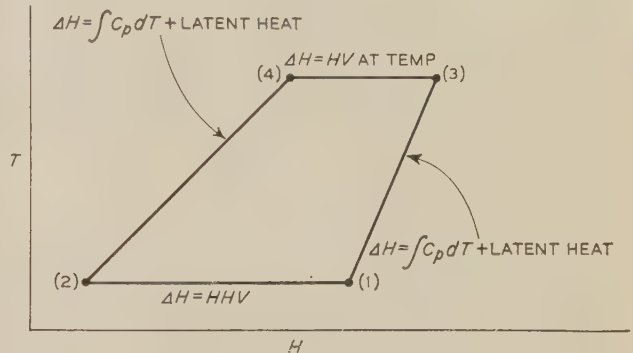


FIG. 1

with some waste products, or both, as often occurs in a gas-turbine cycle, so that the temperature and enthalpy are represented by point (3). If we run our combustion at T_3 and remove the heat of combustion, on cooling to T_2 , we will reach an enthalpy at point (4), such that the heat of combustion or "heating value" at the higher temperature ($H_3 - H_4$) is, in general,¹ different from $H_1 - H_2$. This difference in heating values at the two temperatures is readily computed, as indicated on Fig. 1, as $\Delta H_{1-3} - \Delta H_{2-4}$, where ΔH_{1-3} is the integral of $C_p dT$ plus the latent heat of evaporation of any liquid fuel for the reactants—air and fuel, and ΔH_{2-4} is the integral of $C_p dT$ plus the latent heat of condensation for the products of combustion—nitrogen, CO_2 , water vapor, and excess air.

Consider, for example, the combustion of three different gaseous fuels at 1040 F, as given in Table 1.

TABLE 1 COMBUSTION DATA OF THREE DIFFERENT GASEOUS FUELS

Gas	A	B	C
Nature.....	Hydrocarbon	Coal gas	Producer gas
Composition ^a	(CH_2) _n	60% H_2 ; 40% CH_4	100% CO
High-heat value, HHV, per cu ft.....	790(n)	614	321
Low-heat value, LHV, per cu ft.....	750(n)	534	321
Products of combustion, (cu ft per cu ft of fuel gas):			
CO_2	1.0	0.4	1.0
H_2O	1.0	1.4	...
O_2 consumed.....	1.5	1.1	0.5
ΔH (heat capacity between 90 F and 1040 F):			
Fuel.....	13.17	20.59	18.68
O_2	28.02	20.55	9.34
Total reactants....	41.19	41.14	28.02
CO_2	27.73	11.08	27.73
H_2O	74.00	103.60	...
Total products....	101.73	114.68	27.73
Heating value at 1040 F:			
High-heat value..	790.0	614.0	321.0
ΔH reactants....	+41.2	+41.1	+28.0
ΔH products....	-101.7	-114.7	-27.7
Heat value at temperature.....	729.5	540.4	321.3

^a Composition includes only combustible components, since inerts have the same specific heats during heating and cooling.

¹ Consulting Engineer. Mem. ASME.
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While it must be admitted that in the examples given the lower heating value is a fair approximation (within 3 per cent) to the

heating value at temperature, it can be regarded only as an approximation since it takes into account only one factor, the latent heat of the water vapor. It is also true that we do not have sufficient data to carry out this type of computation accurately. That lack is the concern of the ASME Research Committee on Properties of Gases and Gas Mixtures. We hope to have the required data in the near future.

The effect of using either higher or lower heating values, in place of the true heating value at temperature, is to make the measured thermal efficiency of any engine dependent upon the fuel used to test it. On the other hand, if we attempt to rate fuels on the basis of some heating value at other than room temperature, we will have the relative values of various fuels dependent on the conditions under which they are used. This seems to the author to be where the variation belongs.

ALTERNATIVE HEATING VALUES

We are then presented with three alternative heating values:

1 The high-heating value is a definite physical concept, readily measured as Dr. Fiöck has shown.² It represents the maximum heat that can be obtained from the fuel above room temperature, and is independent of the characteristics of the machinery in which it is used, or of the composition of the fuel. It is a reasonable basis for general comparison of fuels, is the usual basis on which they are sold, and, with additional information as to composition and specific heats of the fuels and their combustion products, it can be used for precise determination of fuel performance under any given conditions.

2 Low-heating value is obtained by computation from the high-heating value, provided the composition of the fuel is known. It is not a true physical concept, but is an arbitrarily defined

approximation to the heating value at temperature to be discussed. It has sanction of wide commercial use in Europe and limited use in this country, as Professor Christie has pointed out.³ The very cogent argument that it gives higher values for thermal efficiency must not be overlooked.

3 The heating value at the working temperature is a precise physical concept which makes the thermal efficiency of any heat engine independent of the nature of the fuel used, provided only that specified temperatures, pressures, and volumes of working fluid prevail throughout the cycle. It, or equivalent calculations, must be used by designers and test engineers for accurate work. For high accuracy it requires a more precise knowledge of the properties of gases and fuels than is now generally available. This information is needed for any accurate work. The heating value at temperature is similar to the low-heating value, but includes all necessary corrections for specific as well as latent heats.

The problem before us is to discuss these three concepts in order that we may clarify and make more accurate our engineering thought and expression. From the purely technical point of view, the author has tried to indicate the soundest theoretical approach to the problem. However, we must also consider carefully established practice, commercial attitudes, and to what extent simplifying approximations are justified by their utility. From this point of view the author is reluctantly forced to suggest that we continue to base general determinations of thermal efficiency on high-heating value, since that is the heat we "pay for," but that we take every opportunity to remind ourselves that the technical comparison of heat engines of all types must be made on the basis of both the machinery and the actual fuel used, these being combined in the concept of heating value at working temperature.

² Refer to page 811 of this issue of the Transactions.

³ Refer to page 819 of this issue of the Transactions.

Symposium on Heating Value of Fuels—

Discussion¹

PAUL DISERENS.² It is possible to define four different calorific values of a fuel, and the conversion of one into another, the high and the low value, at either constant pressure or constant volume, with appropriate experimental procedure. This was done by Cragoe of the Bureau of Standards in 1929.

These values are physical properties of a fuel, and it seems proper that they should be established by the U. S. Bureau of Standards and accepted by engineers, especially for converting measures, weights, or volumes of fuel consumption per horsepower or kilowatt-hour into hourly heat rates, or thermal efficiencies.

The selection of that one of the four possible calorific values and the details of procedure to be used in applying it in a thermal-efficiency calculation might be standardized by the ASME Power Test Code Committee to be used for all sorts of fuels and power-generation equipment, and adopted by all of the Subcommittee individual Codes.

On the other hand, each of the several Subcommittees of the Power Test Code Committee might be allowed or instructed to select that one of the four possible calorific values which it considers most suitable, with the details of procedure in applying it.

Under the first condition, all thermal-efficiency values for all sorts of power-generating plants and any fuel, would have the same basis, and be directly comparable. On the other hand, under the latter condition, those efficiency values which have a different basis would not be directly comparable but the conversion of any one to the equivalent for the basis of the other could be defined.

Thus there is a Test Code policy question to be decided. Is it best to have a single standardized basis for all, or alternatively, is it best to leave each group free to make its own selection, always accepting Bureau of Standards values, definitions, and methods of conversion?

It is the writer's opinion that the Power Test Codes Committee should select one of the four possible calorific values, and that it be adopted by all subcommittees for individual Codes.

The writer prefers the low value because:

(a) It permits direct comparison, as nearly as may be, of the performance of different types of apparatus regardless of the type of fuel used.

(b) It avoids assuming a unique position with respect to the practice in other countries. In Europe the low value is universally used.

On the other hand, if the Committee wishes to leave the matter to the judgment of each individual committee, then it should insist that conversion factors be provided and used so as to permit direct comparison of efficiency values on some selected common basis.

W. F. FRIEND.³ Attention again being directed to practice in expressing the heat value of fuels, and the related discussions centering around certain specialized applications of fuel, indicate the need for perspective and for willingness to accept compromise.

¹ This is a composite discussion presented in connection with the three papers which constitute the present Symposium on Heating Value of Fuels; they appear on pages 811-822 of this issue of the Transactions.

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³ Mechanical Engineer, Ebasco Services Incorporated, New York, N. Y. Jun. ASME.

Too often it is forgotten that three parties are mainly concerned, namely, the suppliers of fuel, the users of fuels and fuel-utilization devices, and the makers of fuel-using equipment. The ASME Power Test Codes for fuels and for thermal equipment embody the practices adopted through the years, in statement of heat values, which best reflect the requirements of all three groups, but particularly the second—the consumers of fuel.

Outstanding as an application of heat-value data is determination of equipment performance and of the fuel component in production costs. Thus for generation of electric power in thermal plants, it is accepted practice to express performance as Btu per kw-hr output, rather than in the form of an output-input ratio or efficiency percentage. Fuel cost is then directly obtained as the product of Btu per kw-hr multiplied by price paid for fuel in cents or dollars per million Btu, with decimal point properly placed. Similar practice exists where mechanical power is produced and for a host of manufacturing operations, in which brake horsepower or units of material turned out become the measure of thermal performance. Fuel, whether solid, liquid, or gas, is universally sold on the high-heat-value basis; performance likewise should be on that basis.

Another general use of production-output to heat-input ratio is in comparing the performance of different plants, equipment types, thermal cycles, kinds of fuel, or operating conditions. For example, the trend in steam power plants with increased pressure, temperature, and reheat, and their standing with respect to internal-combustion-engine plants and the newer gas-turbine cycles with their many variations in heat-exchanger devices, are commonly compared and analyzed on Btu-per-kw-hr basis. Industrial furnaces, which also use huge quantities of fuel, are similarly evaluated, often with direct reference to fuel cost and to possible savings in fuel expense.

Secondary to the interests of the user, which endure through the life of the equipment or of the thermal process, are those of the equipment manufacturer and of the engineer who is the intermediary between manufacturer and user. Both of these should follow practice consistent with that of the customer or client. Equipment designers and research workers face more complex technical problems, for which they may freely devise and employ specialized procedures and terminology, but they should not seek to impose these on others, important though they may seem from the technician's viewpoint. In this category are the low-heat value of gas fuel for internal-combustion engines and the effective heat-value-at-temperature now coming into use for gas-turbine and jet-propulsion apparatus.

In his paper Professor Christie ably summarizes the practices now universal for steam and for internal-combustion power plants operated on liquid fuel, also for steam locomotives and tentatively for gas-turbine power plants—all using the high-heat value. He also points out that solid, liquid, and gas fuels are sold on the basis of high-heat value per unit. The single exception is with respect to gas engines, some designers taking the position that because the hydrogen content of fuel affects the amount of latent heat in the water vapor produced by combustion, it is difficult to relate the test performance with the guarantee, if the hydrogen content of gas used for test differs from that specified in the contract. This difficulty is considered to be eliminated if low-heat value is used instead of high-heat value.

The difficulty encountered with gas engines has been over-emphasized by citing the difference in values as reaching 15 per

cent if the fuel is pure hydrogen. The great majority of gas-fuel engines in use today operate on natural gas, for which a difference of only 8.6 per cent applies to the typical composition in Table 4 of the Power Test Code for Gaseous Fuels, and not over 10 per cent for natural gas from practically all other fields. With blast-furnace gas the differential drops to 1.8 per cent. Compared with these, liquid fuels, ranging from gasoline of 55 API gravity to fuel oils of 25 API gravity, have 7 per cent difference; bituminous coals show between 3 and 4 per cent.

As is evident from the heat-value differentials cited, the same order of magnitude applies in Diesel-engine practice, as for internal-combustion engines with gas fuel. However, for liquid-fuel engines, it is immaterial whether the grade of fuel used in testing is the same as that specified for normal operation, whereas with gas engines there is appreciable difference. Precision in statement of thermal performance can be obtained, adhering to high-heat value of fuel, if the fuel composition is defined with reference to actual value or permissible limit of hydrogen-to-carbon ratio, or if the magnitude of the so-called hydrogen loss is stated.

A serious objection to the use of low-heat value for gas engines is that it "scrambles" inherent limitations of the cycle with performance of the prime mover as a mechanical device. If the gas engine must reject heat above dew-point temperature of its combustion products, and therefore is unable to utilize latent heat of water vapor, neither can the Diesel. Steam cycles similarly are limited, whether using natural gas or other fuels having hydrogen content. Moreover, latent heat in combustion products is not the only unavailable component; heat in the exhaust steam and velocity energy of steam clearing the last row of turbine blades also are inescapable losses. It would be inconsistent to make adjustment for the one loss and to ignore the others.

The dilemma, if such it be, might be dealt with rationally by means of cycle evaluation like the "Rankine cycle efficiency" used in days of the steam engine and later carried over into turbine practice. Performance of the actual engine could be compared to that of the ideal internal-combustion cycle taking into account compression ratio, exhaust-gas temperature, and other factors for which the datum values would be agreed upon by Test Code or other authority. However, the Rankine-efficiency concept has long since outlived its usefulness, and with the present rapid progress in many forms of the internal-combustion cycle, the writer has grave doubt as to advisability of further complicating the terminology and procedure.

Expression of heat rate for certain types of prime mover, on the basis of fuel heat value different from that used for other types and by industry generally, has led to misunderstandings in the past. It would be regrettable if these should be allowed to continue through lack of standardization, where standardization is desirable and feasible. Further efforts to reconcile procedures should be made, particularly with respect to the ASME Power Test Codes.

L. H. Fry.⁴ The three papers, and to a still greater degree the discussion, show that the term "heating value" may be made to cover a multitude of concepts. The scientists may split hairs and slice the ideas very thin, but as engineers we should ask for simple definitions which can be used in our work. Professor Christie's quotations from the Codes show that these are on firm ground.

Dr. Kleinschmidt's concluding paragraph points out that the technical comparison of heat engines of all types must be made on the basis of both machinery and fuel. This deserves strong emphasis, and appeals to a locomotive engineer.

In comparing the performance of steam and Diesel locomotives, a point is often made of the fact that the Diesel has a thermal

efficiency of 24 per cent against 7 per cent for steam. This may be true, but the original thermal units fed to the engine are purchased in the form of oil in one case, and in the form of coal in the other. If a Btu in oil costs 3.25 times as much as a Btu in coal, there will be very little difference in the cost of the Btu's which appear at the locomotive drawbar.

JOSEPH KAYE.^{5,6} Let us consider any power plant which operates in an atmosphere of constant pressure and temperature. The power plant receives streams of fuel and air at the temperature and pressure of the atmosphere and rejects products of combustion which must attain ultimately the pressure and temperature of the atmosphere. The values of the pressure and temperature of this atmosphere will vary with the position above or below the surface of the earth. The power plant under consideration is then a general one and includes, practically speaking, all power plants ranging from steam and gas-turbine plants, jet-propulsion and rocket devices to submarine power plants.

There are two questions which may be raised by those interested in a figure of merit for such power plants:

1 Is there any "rational" basis for comparison of diverse power plants if each operates with the same mixture of fuel and oxidant?

2 Is there any rational basis for comparison of various fuels and oxidants which are used in power-producing devices?

It is shown that the answer to each of these questions is provided by the use of the decrease in free energy of the system of fuel and oxidant.

Let us begin by noting that the efficiency of a power plant is an arbitrary figure of merit. It is usually defined as the ratio of the net work output to a particular property of the fuel-oxidant system used in the power plant. Among the many properties of the fuel-oxidant system which have been used as the denominator of this ratio are "higher heating value," "lower heating value," "enthalpy of combustion," "heat of combustion," "net heat of combustion," etc. Some of these terms are used often without precise definition with respect to states of the reactants and products. Even when they are adequately defined, alas, the significance of the corresponding efficiencies is not evident. Furthermore, a comparison of the efficiencies of two power plants using different fuels is of dubious rationality and utility.

The efficiency of a power plant may be measured rationally by means of the availability or the free energy of a system. Let us define efficiency as the ratio of the net work output to the "maximum useful work" which could be obtained from the fuel-oxidant system while it exchanges heat only with the atmosphere. This maximum useful work is the decrease in free energy⁸ or the decrease in availability⁹ of the fuel-oxidant system between the initial state of pressure and temperature equilibrium with the atmosphere and its most stable state in the atmosphere.⁹ As regards availability, the base temperature and pressure are con-

⁵ Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Jun. ASME.

⁶ This discussion is based neither on new thermodynamic functions nor on new applications of these functions. In fact, the thermodynamic functions used here, namely, the free energy and the availability, were invented nearly 70 years ago by J. W. Gibbs. Since then several authors⁷ have shown the utility of these functions in the analysis and in the understanding of engineering problems.

⁷ "Thermodynamics," by J. H. Keenan, John Wiley & Sons, Inc., New York, N. Y., 1941.

⁸ "The Rational Definition of Steam Turbine Efficiencies," by G. Darrieus, *Engineering*, vol. 130, 1930, pp. 283-285.

⁹ Ibid., J. H. Keenan.

⁹ Because it is generally an uneconomical objective, work obtainable from diffusion of the products into the atmosphere is ignored in this statement. The fuel-oxidant system is always considered to be separate and distinct from the atmosphere.

⁴ Fellow ASME. Deceased July 10, 1948.

sidered to be those of the atmosphere. This maximum useful work could be realized in a completely reversible power plant which operates in an atmosphere of constant temperature and pressure. It should be noted that this definition is not restricted to the performance of a given machine or device but is given in terms of the properties of the fuel-oxidant system which changes its state chemically in surroundings at constant temperature and pressure.

The maximum useful work may be calculated readily from thermodynamic data on fuel-oxidant systems by noting that the decrease in free energy is given by

$$F_R - F_P = H_R - H_P - T(S_R - S_P)$$

where F , H , S , and T denote the free energy, enthalpy, entropy, and thermodynamic temperature, respectively, and subscripts R and P refer to reactants and products, respectively. Thermodynamic data are now available for many hydrocarbon fuels through the efforts of Rossini and his co-workers at the National Bureau of Standards.

The rational definition of efficiency just presented is superior to the older definition for many reasons. (1) It includes the restrictions imposed on processes by the second law of thermodynamics through comparison of the actual work output of a device with the maximum useful work of a reversible device operating between the same states of the fuel-oxidant system. (2) For a given mixture of hydrocarbon fuel with air, the maximum useful work is given with sufficient precision for engineering purposes by a single number, not by eight different numbers as is the case for constant-pressure heat of combustion. This distinction may be illustrated by calculation of the decrease in enthalpy or in internal energy for the reaction at constant temperature and pressure, (using the recent data of Rossini and co-workers) for a typical fuel such as octane

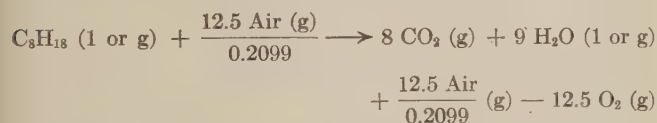


TABLE 1

Reactants	Products	$H_R - H_P$ Btu per lb-mole	$E_R - E_P$, Btu per lb-mole
Octane (l)	H ₂ O (l)	2,352,000	2,347,000
Octane (l)	H ₂ O (g)	2,182,000	2,187,000
Octane (g)	H ₂ O (l)	2,370,000	2,364,000
Octane (g)	H ₂ O (g)	2,200,000	2,204,000

Table 1 shows that selection of either a high- or a low-heating value is not justifiable on a rational basis. The decrease in free energy for the foregoing system, however, is given within 0.1 per cent by a single number, 2,269,000 Btu per lb-mole of octane, for all four possible states of the fuel-air system, i.e., for either liquid octane or gaseous octane at the vapor pressure of the liquid in the reactants state, for liquid water or for water vapor at the vapor pressure of the liquid in the products state.

A third advantage of the rational definition of efficiency, of a practical nature, is that the decrease in free energy for many hydrocarbon fuels is approximately the average of the high and low "heats of combustion," defined as the decrease in enthalpy at constant pressure and temperature of liquid octane and air to form products with liquid water and gaseous water, respectively. This means that the present system of numbers expressing efficiencies of power plants will not suffer a marked change in magnitude through introduction of the new efficiency. However, it should be noted carefully that this approximate equality of the decrease in free energy and of the decrease in enthalpy does not hold for many fuels or for many chemical reactions; in fact, Parks

and Huffman¹⁰ point out that many chemical reactions exist where the decrease in free energy is of the opposite sign to the decrease in enthalpy.

The fourth advantage of this rational definition of efficiency is that it may be inverted readily to provide a figure of merit for a work-absorbing device which carries out a chemical reaction. For example, in a power-absorbing device, whose purpose is the separation of air into its components, a rational figure of merit is the ratio of the "minimum work input" to the actual work input required to start with air and produce oxygen and nitrogen (ignoring the rare gases) at the same pressure and temperature. The minimum work input is the decrease in free energy of the air from its initial state to oxygen and nitrogen at the same pressure and temperature. Thus the figure of merit, based upon the decrease in free energy or availability at constant pressure and temperature is a general one, applicable to both work producers and work absorbers.

With regard to the question (2), various systems of fuels and oxidants may be compared thermodynamically by means of the "maximum useful work," which could be obtained from the fuel-oxidant system while it exchanges heat only with the atmosphere. In other words, the decrease in free energy may be used as a rational basis of comparison of various fuel-oxidant systems. Any such comparison is evidently a purely thermodynamic one which omits from consideration questions of economics and convenience.

HERBERT KUENZEL.¹¹ In discussing the matter of heats of combustion of fuels and their use in computing thermal efficiencies, it seems to the writer to be more accurate, although more inconvenient, to use the chemical energy of the fuel in preference to either the so-called higher or lower heat of combustion of that fuel.

Indeed, much of the confusion remarked by the authors of these papers in the use of heats of combustion may have arisen from the fact that heats of combustion actually are not the energy supplied to a utilizer, but are merely convenient terms arrived at by test in which the corrected heat transferred to water is taken as a proper denominator in the efficiency expression.

If accuracy of concept and also of numerical result is the goal, it would seem desirable to dispense with "heats of combustion" and use the more fundamental concept of the chemical energy of the fuel. The chemical energy of a fuel can be computed from the same sort of a test that is used to determine the heat of combustion; and while, as Dr. Kleinschmidt points out, data may not as yet be available to make this computation accurately, we probably should revise our thinking on this point and await the time when the ASME Research Committee on Properties of Gases and Gas Mixtures will have remedied the lack.

W. Y. LEWIS.¹² In determining by the Mahler system the higher calorific value of a fuel with or without moisture, the products of combustion are assumed to take the initial surrounding temperature, and the steam from the water coming either from the combustion of the hydrogen existing in the fuel, or from the moisture, is supposed to be entirely condensed.

In the case of lower calorific value, the steam existing in the gaseous products of combustion is assumed not to be condensed although brought back to the temperature of the surrounding medium. Therefore a deduction from the higher calorific value

¹⁰ "The Free Energies of Some Organic Compounds," by G. S. Parks and H. M. Huffman, Chemical Catalog Company, New York, N. Y., 1932.

¹¹ Associate Professor of Mechanical Engineering, Washington University, St. Louis, Mo. Mem. ASME.

¹² Fairmont Coal Bureau, St. John's Wood, London, W.W.8, England.

has to be made to moisture and to combustion of hydrogen in the coal.

It is the lower calorific value of the green fuel that should be debited against the boiler, since the heat due to the combustion of the hydrogen forming steam in the products and the heat of the steam due to the moisture are not recoverable in the boiler as in the calorimeter.

E. S. DENNISON,¹³ To compute the heating value of a fuel at an elevated temperature, Dr. Kleinschmidt assumes that both fuel and air have been preheated to the specified temperature before the reaction takes place. A perfectly valid definition of the heating value at temperature results from this assumption, and the author has made the necessary calculations for several gaseous fuels. However, it is not clear what he would do in the case of a liquid or solid fuel, oil or coal, which suffers physical and chemical changes if separately heated to a high temperature. It is not easy to assign a value of enthalpy to the fuel in that case. Also, it is questionable whether the heating value so defined would be convenient for performance calculations, except perhaps for a reciprocating gas engine.

It is common in combustion power cycles, for example, gas-turbine cycles, to preheat the air but to introduce the fuel at or near room temperature. In this case also, there is a deficiency in lower heating value as compared to that determined from a laboratory test at room temperature, and the amount of the deficiency is important for design and performance calculations. This case is so frequently encountered as to be of general interest. Fortunately, the existing data permit the low-heat-value deficiency to be estimated closely.

The equation giving the effective LHV for 1 lb of fuel at temperature t may be written

$$(\text{LHV})_t = (\text{LHV})_o + W_o \overline{c_{po}} (t - t_o) + c_f (t_f - t_o) - W_{cpc} \overline{c_{pc}} (t - t_o) - W_{spc} \overline{c_{ps}} (t - t_o)$$

in which

$(\text{LHV})_t$ = effective heating value at temperature t

$(\text{LHV})_o$ = lower heating-value at room temperature, usually at 68 F

W_o, W_c, W_s = weights of O_2 , CO_2 , and H_2O , respectively

$\overline{c_{po}}, \overline{c_{pc}}, \overline{c_{ps}}$ = mean constant-pressure specific heats of O_2 , CO_2 , and H_2O , respectively, between t_o and t

c_f = specific heat of fuel

t = air temperature deg F

t_o = room temperature (68 F)

t_f = fuel temperature, deg F

In most cases the amount of heat represented by $c_f (t_f - t_o)$ can be neglected. The loss of effective heating value is then

$$\Delta (\text{LHV}) = W_o \overline{c_{po}} (t - t_o) + W_s \overline{c_{ps}} (t - t_o) - W_c \overline{c_{pc}} (t - t_o)$$

Here the weight W_o of oxygen is that required to combine with the carbon in the fuel to form CO_2 , plus that to combine with the hydrogen to form H_2O . By inspection, it is evident that the hydrogen and carbon present in the fuel act independently to diminish the effective heating value. Therefore it is useful to compute $\Delta (\text{LHV})$ at various temperatures for pure hydrogen and for pure carbon. This has been done and the results are given in Fig. 1 of this discussion. The deficiency for hydrogen is roughly 10 times its value for carbon, and the curves have been plotted to different scales. The large effect in the case of hydrogen is due to the high specific heat of steam, and to the fact that about 9 lb of it are formed by the combustion of 1 lb of H_2 .

¹³ Director of Research and Development, Elliott Company, Jeanette, Pa. Mem. ASME.

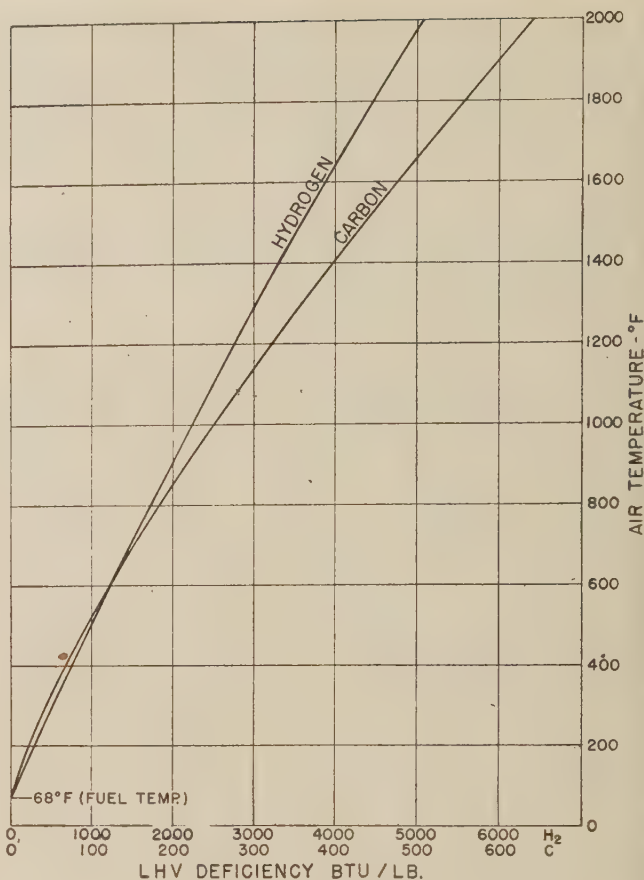


FIG. 1 CHART FOR ESTIMATING EFFECTIVE HEATING VALUES FOR HYDROCARBON FUELS

Specific-heat data for these computations were taken from previous work.^{14,15}

Having constructed these curves, it is a simple matter to estimate the LHV deficiency for any actual fuel. For example, at 1040 F, $\Delta (\text{LHV})$ for C is 265 Btu per lb and for H_2 it is 2350 Btu per lb.

No. 2 Fuel: 86.5 per cent C, 12.6 per cent H, 0.9 per cent inerts

$$0.865 \times 265 = 229$$

$$0.126 \times 2350 = 296$$

$$\Delta (\text{LHV}), \text{ Btu per lb} = 525$$

Methane CH_4 : 75 per cent C, 25 per cent H

$$0.75 \times 265 = 199$$

$$0.25 \times 2350 = 588$$

$$\Delta (\text{LHV}), \text{ Btu per lb} = 787$$

In the case of Fig. 2 fuel, the loss of heating value at 1040 F is about 2.9 per cent, while for methane it is 3.65 per cent. The deficiency in the latter case is of course greater than it would be according to the author's assumptions.

A heavy fuel oil may be preheated, for example, to 250 F preparatory to burning it. The heat represented by $c_f (t_f - t_o)$ is then about $0.475 (250 - 68) = 86.5$ Btu. This represents about 0.005 $(\text{LHV})_o$ for that fuel. Hence even in this extreme case for a liquid fuel, the effect of fuel preheat is hardly significant.

¹⁴ Cornell University Engineering Experiment Station, Bulletin No. 30, October, 1942.

¹⁵ "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, John Wiley & Sons, Inc., New York, N. Y., 1936.

The chart, Fig. 1, is offered as a convenient device for estimating effective heating values for all hydrocarbon fuels, particularly for gas-turbine-cycle calculations assuming the LHV at room temperature to be known.

L. C. LICHTY.¹⁶ It seems obvious that in any single engine test there is a definite thermal efficiency indicating the part of the energy supplied which has been transformed into shaft work. Consequently, regardless of the procedure of evaluation, the thermal efficiency should be the same if each procedure is correct.

Fiock calls attention to the possibility of calculating any one of the usual four heats of combustion from any other by applying certain modifications which relate to the various calorimetric processes. Thus all of the four usual heats of combustion are related uniquely, and thermal efficiencies of engines can just as well be determined with any one of the heats of combustion. This does not mean that the heat of combustion should be the sole term in the denominator of the efficiency expression but that if the thermal efficiency of the engine is evaluated correctly, the efficiency determined should be the same, irrespective of the particular heat of combustion used in its determination.

The writer and others contend that it is the fault of the power plant rather than the fuel if it does not use all the energy available. However, even though the writer uses the high-heat value he does not agree that the high or any other heat of combustion should necessarily define the energy supplied to a combustion process which uses the same fuel and oxygen but occurs usually under other than calorimetric conditions.

For the case, outlined by Kleinschmidt, of air which is preheated or compressed to a temperature appreciably above the ambient condition, it seems quite obvious that more energy has been supplied to the process than if the air were supplied at the ambient temperature. Also, it seems obvious that if the engine is operating in the atmosphere, and is unable to reduce the products of combustion to the ambient temperature in its work process, it is the fault of the machine. In all cases, the energy supplied should be charged against the machine, and this is usually not the heat value of the fuel.

Professor Christie states that low-heat values are used in computations of flame temperatures and other calculations "for which the high-heat value does not apply." Evidently this is incorrect since the low-heat value is computed from the high-heat value and consequently either value may be used for such computations. Thus for adiabatic, steady-flow combustion processes with negligible kinetic-energy effects, or for constant-pressure combustion, in which H is enthalpy and M is quantity of medium

$$(\text{LHV})_{T_1} = M_{\text{prod}}(H_{T_2} - H_{T_1})_{\text{prod}} \dots \dots \dots [1]$$

all products being gaseous; or

$$(\text{HHV})_{T_1} = M_{\text{prod}}(H_{T_2} - H_{T_1})_{\text{prod}} \dots \dots \dots [2]$$

the H_2O being liquid at T_1 only.

Subtracting Equation [2] from Equation [1] results in

$$(\text{HHV} - \text{LHV})_{T_1} = M_{\text{H}_2\text{O}} (H_{\text{gas}} - H_{\text{liq}})_{T_1} \dots \dots \dots [3]$$

which is the definition of the difference between low- and high-heat value. Thus Equation [2] may be used as well as Equation [1].

It should be noted that using heat values in Equations [1] and [2] is resorting to fictitious processes of heating products from T_1 , at which temperature they never exist in the actual combustion process.

¹⁶ Professor of Mechanical Engineering, Yale University, New Haven, Conn. Mem. ASME.

In 1930, Keifer and Stuart¹⁷ proposed a method which the writer has adopted,¹⁸ which simplifies the analysis of combustion processes and treats them as they occur and not as fictitious processes, although the same result is obtained if the computations are made correctly both ways. Writing the energy equation for the ideal bomb-calorimetric process

$$C_{f+O_2} + (U_f)_T + (U_{O_2})_T = (U_{\text{prod}})_T + (\text{HHV})_{T,V} \dots \dots [4]$$

in which C is the chemical energy associated with the reaction, it being a constant for a given fuel-oxygen reaction; U is the internal energy of the various constituents alone, before and after the reaction at temperature T ; HHV is the high-heat value for the fuel at the specified conditions, varying with temperature of process as indicated by subscript T ; f and a indicate fuel and air, respectively.

Item C may be evaluated from energy Equation [4] if the heat value and internal-energy values for the various constituents are known above some arbitrary datum plane and may be used in any combustion process for the given fuel, provided the same internal-energy tables are used. C is the heat value at absolute zero temperature if the internal-energy values are on an absolute basis as indicated by Prof. G. A. Goodenough¹⁹ in 1911. Using a datum plane of 520 deg R for zero internal energy, C is the heat value at absolute zero plus (and minus) the constants of integration to make the U values in Equation [4] absolute at 520 deg R.

Having computed the C value for any fuel from the heat value and energy data, the analysis of any combustion process is accomplished by writing the energy equation for the given process. For the adiabatic steady-flow burner process with negligible kinetic-energy effects, the fuel and air flowing into the burner supply the chemical energy of the fuel-oxygen reaction as well as the enthalpy of the fuel and air at their respective entering temperatures, while for complete combustion all of this energy appears as enthalpy of the products which includes the excess air, at the final temperature. Thus

$$C_{f+O_2} + (H_f)_{T_f} + (H_a)_{T_2} = (H_{\text{prod}})_{T_2} \dots \dots \dots [5]$$

in which the various temperatures on the left-hand side of the equation may be appreciably different from the standard temperature for calorimetric determinations.

Heat value does not appear in this relation, since it is not an energy term in any energy equation for any process except the calorimetric process. However, it can be manipulated into the energy equation for any combustion process by combining the energy equations for the process with the energy equation for the calorimetric process at the same fuel and air temperature of the actual process. Thus for the initial conditions in Equation [5] the energy equation for the calorimetric process is

$$C_{f+O_2} + (H_f)_{T_f} + (H_a)_{T_2} = (H_{\text{prod}})_{T_2} + (\text{HV})_{T_f \text{ and } T_2} \dots \dots [6]$$

Combining Equations [5] and [6] results in

$$(\text{HV})_{T_f \text{ and } T_2} = (H_{T_3} - H_{T_2})_{\text{prod}} \dots \dots \dots [7]$$

Thus we arrive at an equation for a fictitious process of heating the products from T_2 , at which temperature they never exist in the actual process, to T_3 , the correct temperature. (HV) in Equation [6] may be either HHV or LHV, depending merely upon whether the $(H_{\text{prod}})_{T_2}$ is evaluated for liquid or gaseous H_2O , respectively.

¹⁷ "Principles of Engineering Thermodynamics," by P. J. Keifer and M. C. Stuart, John Wiley & Sons, Inc., New York, N. Y., 1930.

¹⁸ "Internal Combustion Engines," by L. C. Lichty, McGraw-Hill Book Company, Inc., New York, N. Y., 1939.

"Thermodynamics," by L. C. Lichty, McGraw-Hill Book Company, Inc., New York, N. Y., 1936 and 1948.

¹⁹ "Principles of Thermodynamics," by G. A. Goodenough, Henry Holt & Company, New York, N. Y., 1911.

The heat value in Equations [6] and [7] is what Kleinschmidt proposes to use in the engine thermal-efficiency determination. It is the correct value to use for computing the maximum temperature attainable by the method indicated in Equation [7] but not necessarily the correct value for the efficiency determination. But why compute the (HV) at various values of (T_2) and (T_f) for each different test when one determination of C for the given fuel permits the computation of maximum temperatures for all cases with that fuel by the actual rather than a fictitious process?

Recently, Newman A. Hall²⁰ made use of the chemical energy as herein defined by introducing correctly into his energy equation for the combustion process the four terms from the calorimetric equation (that algebraically summed are equal to C) instead of evaluating C and eliminating HV and the other three terms.

Considering only turbine thermal-efficiency determinations, it seems illogical to use (HV) at (T_2) and (T_f) as the basis for determination of efficiency in Kleinschmidt's case. The mere fact that the air is supplied at higher than ambient temperature indicates that the turbine only should be charged with more than the energy liberated by the fuel. The energy equation for the elementary gas-compressor-turbine process, neglecting kinetic energy effects, is

$$(\dot{H}_a)_{T_2} + (\dot{H}_f)_{T_f} + W_{\text{comp}} + C_{f+O_2} = W_{\text{turb}} + (\dot{H}_{\text{prod}})_{T_4} + Q_{\text{rad}} \dots \dots \dots [8]$$

in which W represents shaft work and Q is the heat radiated or transferred from the burner and turbine. The sum of the first and third terms is equal to $(\dot{H}_a)_{T_2}$, the enthalpy of the air leaving the compressor if the process is adiabatic.

The efficiency of the turbine alone is the ratio of the work obtained divided by the energy supplied. Equation [8] indicates that the energy supplied is the chemical energy associated with the reaction plus the enthalpy of the entering media above that associated with the ambient condition. Thus

Turbine thermal efficiency

$$= \frac{W_{\text{turb}}}{C_{f+O_2} + (\dot{H}_a)_{T_2} + (\dot{H}_f)_{T_f} - (\dot{H}_a + \dot{H}_f)_{T_a}} \dots [9]$$

in which $(\dot{H}_a + \dot{H}_f)_{T_a}$ is the enthalpy of the air and fuel at ambient conditions. Equation [9] may be written with heat value in the following manner

Turbine thermal efficiency

$$= \frac{W_{\text{turb}}}{(HV)_{T_2 \text{ and } T_f} + (\dot{H}_{\text{prod}})_{T_2} - (\dot{H}_a + \dot{H}_f)_{T_a}} \dots [10]$$

If $T_f = T_2$, and if the enthalpy value is zero for all constituents at the ambient temperature T_a , the turbine should be charged with appreciably more energy than the heat value at T_2 . In the limiting case of no fuel burned, the turbine only would do some work, but the efficiency of the turbine only based on Kleinschmidt's proposal would be infinite. Thus it is obvious that the turbine only should be charged with the enthalpy of the air above the ambient condition as indicated in Equation [9].

For the efficiency of the combined turbine and compressor process

Over-all efficiency

$$= \frac{W_{\text{turb}} - W_{\text{comp}}}{C_{f+O_2} + (\dot{H}_a)_{T_1} + (\dot{H}_f)_{T_f} - (\dot{H}_a + \dot{H}_f)_{T_a}} \dots [11]$$

or

Over-all efficiency

²⁰ "The Fuel-Air Ratio Required for Constant Pressure Combustion of Hydrocarbon Fuels," by N. A. Hall, digest printed in *SAE Journal*, vol. 54, December, 1946, pp. 32-36.

$$= \frac{W_{\text{turb}} - W_{\text{comp}}}{(HV)_{T_1 \text{ and } T_f} + (\dot{H}_{\text{prod}})_{T_1} - (\dot{H}_a + \dot{H}_f)_{T_a}} \dots [12]$$

Thus it seems obvious that the denominator in the efficiency formula should not be any particular heat value only, even though it is usually equivalent to the major portion of the energy supplied, but that whatever heat value is used should be corrected to make the denominator the total energy supplied (including the enthalpy of the mediums above ambient conditions), and this should result in the same efficiency regardless of which heat value is used. Also, this analysis is free from fictitious processes when the chemical-energy term, which belongs in every energy equation for combustion processes, is used.

For some time it has been the custom to set up tables of internal energy or enthalpy of gases including H_2O above some arbitrary datum plane such as 60°F at which the internal energy in the gaseous state is assumed to be zero. For a gaseous fuel such as CH_4 , in which the moles of air and fuel are equal to the moles of products formed in the calorimetric process, the $\dot{H}_{\text{prod}}$ will equal the $(\dot{H}_a + \dot{H}_f)$ at the datum temperature. Then, assuming that the various T values in Equation [12] are the same and equal to the standard temperature at which calorimeter determinations are reported, and that this is the datum temperature for the enthalpy values of the gaseous substances, the denominator is equivalent to the low-heat value of the fuel.

If the enthalpy datum plane is absolute zero temperature, the denominator for the efficiency term at 540 deg R is greater than the low- and high-heat values by the following percentages for the given gaseous fuels²¹

Fuel	CO	H ₂	CH ₄	C ₄ H ₁₀	C ₆ H ₁₄
LHV...	3.2	16.3	5.6	10.8	10.4
HHV...	...	-1.55	-4.8	2.3	2.4

With an absolute temperature datum plane for energy values toward which we are going, it becomes quite apparent that the term $(\dot{H}_a + \dot{H}_f)_{T_a}$ in the denominator of the efficiency expression is an important term which tends to eliminate the effect of the present arbitrary energy datum plane on efficiency.

In conclusion, the confusion that exists in the minds of many regarding heat value and thermal-efficiency determinations appears to be the result of an endeavor to apply the cycle analysis to a process which is not a cycle. In a cycle, the medium starts from and returns to the original state after a series of processes. In such cases it is quite obvious how much heat is added from a hot source, and there is no disagreement regarding the denominator of the efficiency expression for a heat-engine cycle. However, the combustion process eliminates the possibility of a cycle, since the products are not and cannot be returned to fuel and air. In fact, no heat is added in the adiabatic combustion process; chemical energy is liberated by the reaction and appears as internal energy or enthalpy of the products. Consequently there is no heat term to be used in the efficiency determination of the combustion gas turbine or internal-combustion-engine process.

Therefore it appears that we shall always have confusion in this matter until all recognize that we are not dealing with a heat-engine cycle, and instead apply some concept of energy supplied to a process such as is indicated by the energy equation for the actual process. Then the ridiculous situation of different thermal

²¹ Data for these fuels and for the computations were obtained from the following sources:

"The New Specific Heats," by R. C. H. Heck, *Mechanical Engineering*, vol. 62, 1940, p. 9.

"Thermodynamics of Gaseous Paraffins," by K. S. Pitzer, *Industrial and Engineering Chemistry*, vol. 36, 1944, p. 829.

"Heats of Combustion and Formation of the Paraffin Hydrocarbons at 25°C," by E. J. Prosen and F. D. Rossini, National Bureau of Standards. *Journal of Research*, vol. 34, 1945, p. 263.

efficiencies with different heat values for a given engine test will be ended.

CLOSURE BY A. G. CHRISTIE

The foregoing discussion has shown that there are many standards by which the heating value of fuels may be measured. Some of these methods are in use for various commercial and research purposes. Others are at the moment of academic interest and are not generally known to those concerned with the commercial testing of power machinery.

The purpose of the Power Test Codes is to establish certain standards and rules which can be understood readily and followed by those who buy, sell, and test power equipment. This is the group for whom the codes are prepared. In establishing standards of measurement, consideration must be given to the points of view of these persons, their familiarity with thermodynamic theory, the ease of application of the test methods, and the commercial customs in industry. These considerations may rule out some of the standards of an academic nature which have been suggested.

The business of the Power Test Codes Committee is to fix certain so-called "yardsticks" as measures of the performance of power machines. These measures of performance must be recognized in the profession as reasonable and fair and determinable by one skilled in the art of testing such commercial equipment. In the choice of such a yardstick, custom and practice receive full consideration. It is not of great importance which of the several recognized standards is chosen so long as it is accepted by the profession as the yardstick for the particular machine. In dis-

cussing or comparing performances, all then refer to the same unit of measurement.

The Power Test Codes Committee is indebted to all who have contributed their discussion as their comments will aid in the selection of standards for the heating value of fuels. The discussion also aids in directing attention to some of the less well-known measures of heat availability that will aid the thinking of many engineers on problems of heat generation outside of commercial tests.

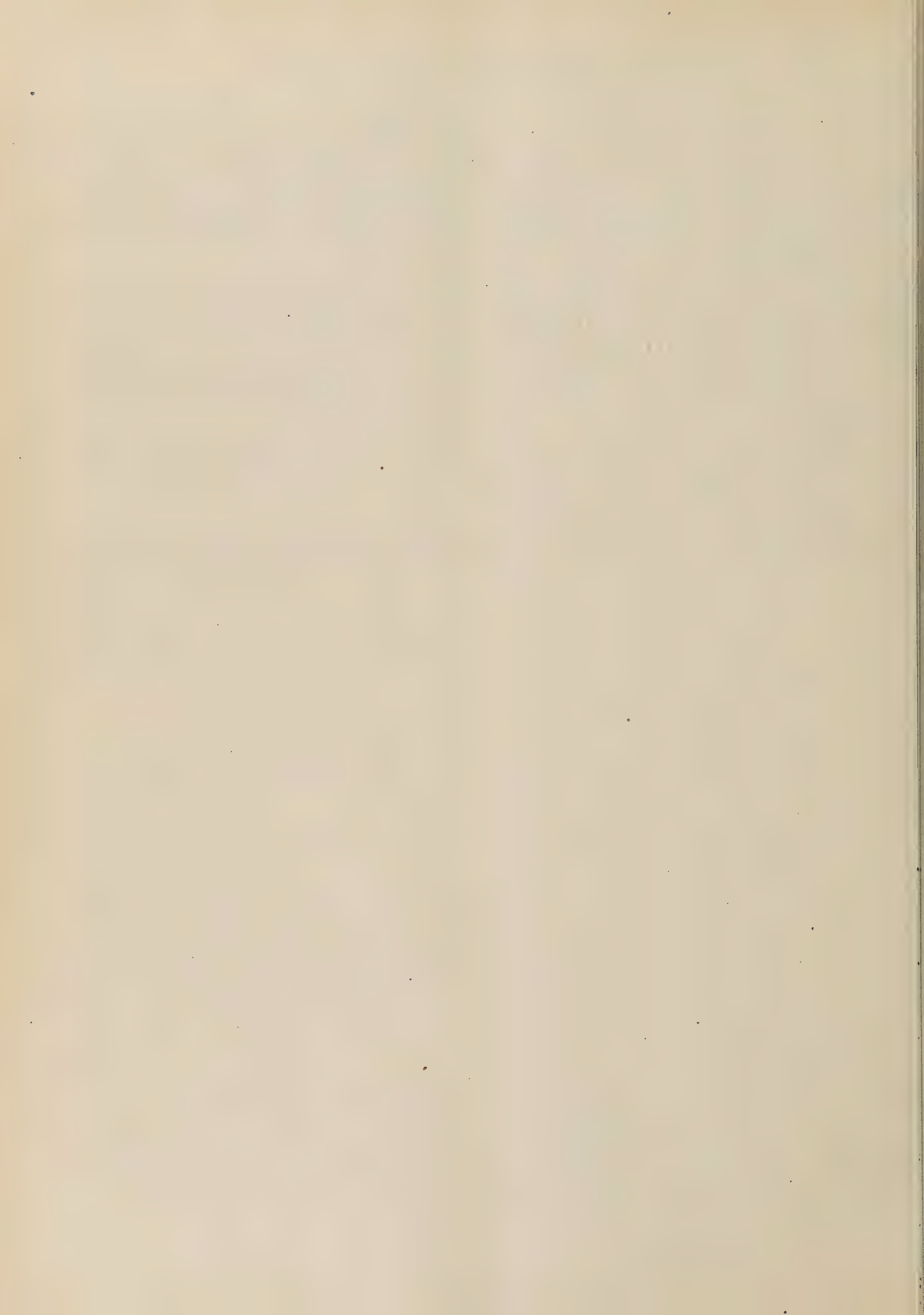
CLOSURE BY R. S. JESSUP

Referring to discussion by Paul Diserens, it is not entirely clear what is meant by the statement that heats of combustion of fuels "should be established by the Bureau of Standards." This Bureau does research on, and makes determinations of heats of combustion for other government agencies only. In some cases data on heats of combustion of certain classes of fuels have been correlated^{22,23} with other properties which can be measured more readily.

The suggestion of Professor Kaye that the decrease in free energy is a better criterion for comparing power plants than heats of combustion is interesting and sound in principle. The requirement that the composition of the fuel be known would impose difficulties in many cases.

²² "Thermal Properties of Petroleum Products," by C. S. Cragoe, NBS Miscellaneous Publication No. 97, 1929.

²³ "Net Heat of Combustion of AN-F-28 Gasoline," by R. S. Jessup and C. S. Cragoe, NACA Technical Note No. 996, 1945.



Silicone Rubber—New Properties for Design Engineers

By G. S. IRBY, JR.,¹ WYMAN GOSS,² AND J. J. PYLE³

Silicone rubber is a new synthetic elastomer which exhibits exceptional properties of thermal stability. The physical, chemical, and electrical properties of this material, as affected by high and low temperatures, have been studied. The results of this study, together with fabrication techniques and proven applications, are compared to show where silicone rubber may be of best use to the design engineer.

AS a result of rapid developments in the field of organo-silicon chemistry during the last few years, a whole new class of high polymeric materials has been made available to product and design engineers. This group includes such products as silicone rubber, oils, resins, greases, and water-repellant films. The members of the silicone family naturally vary as to properties and potential applications. However, they all offer an outstanding characteristic, namely, thermal stability from -70°F to 500°F .

Silicone rubber, the elastomeric member of this family, shows its temperature stability by not only retaining its flexibility, resiliency, and surface hardness over this entire temperature range but also by resistance to long periods of heat-aging. The availability of an elastomer with these heat-resistant properties immediately suggests new design possibilities where elastic materials are needed in equipment operating at elevated temperatures. However, mistaken conceptions of this material will arise if it is merely substituted for conventional rubbers in existing designs. The properties of silicone rubber do not match those of natural or synthetic rubber at room temperature. On the other hand, at operating temperatures of -70°F , or at 500°F , the properties of other elastomers generally fall far short of silicone rubber. It is the purpose of this paper to discuss these properties with related engineering applications to show where silicone rubber can and cannot be used.

FORMATION AND PROCESSING

Silicone rubber is produced by compounding a silicone gum with suitable fillers and then vulcanizing or cross-linking the compounded rubber stock. However, to produce the gum, a considerable amount of chemistry is involved.

The gum is produced commercially in two different ways, Fig. 1. Through a direct reaction, methyl chloride and silicon are combined in the presence of copper as a catalyst to form a mixture of methyl chlorosilanes. This mixture is fractionated to obtain dimethyl dichlorosilane which in turn is hydrolyzed to produce a silicone oil. This partially condensed methyl-silicone oil is then further polymerized to produce the final gum product.

In the Grignard process, silicon tetrachloride, methyl chloride,

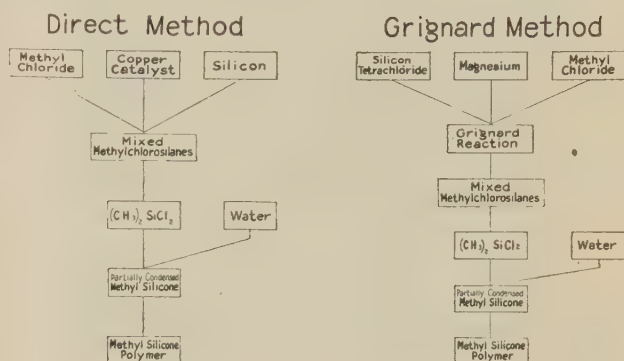


FIG. 1 COMMERCIAL PRODUCTION OF DIMETHYL SILICONE POLYMER

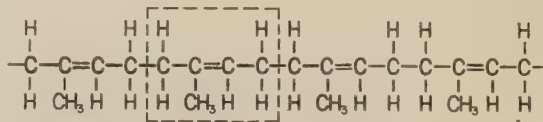


FIG. 2 THE POLYMER OF NATURAL RUBBER

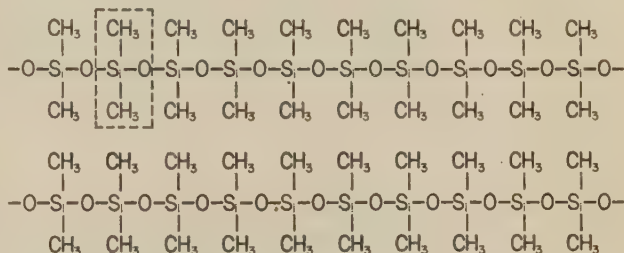


FIG. 3 THE POLYMER OF SILICONE RUBBER

and magnesium are starting reactants. Through a Grignard reaction a mixture of methyl chlorosilanes is formed, and from this point the procedure can be the same as in the direct process to produce a methyl-silicone gum.

A molecule of silicone gum compared to one of natural rubber, Figs. 2 and 3, illustrates the difference between the two substances. The fundamental structure of the natural-rubber molecule is a carbon-to-carbon linkage, while in the silicone molecule, a silicon-oxygen linkage is the basic structure. It is this particular structure that imbues silicone rubber with its unique properties of thermal stability.

From the gum, compounded rubber stocks are produced by milling or banburying with such fillers as calcium carbonate, lithopone, titania, and various silica fillers. Carbon black has not been used to a great extent in silicone rubber to date because of vulcanization difficulties when this particular filler is used. The hardness of the stock is a function of the type of filler, filler-gum ratio, and degree of vulcanization. The filler-to-gum ratio can be varied extensively to secure desired properties. Excellent rubbers can be compounded using filler-to-gum ratios as high as 2 to 1.

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Contributed by the Rubber and Plastics Division and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-126.

It has not been found possible to vulcanize silicone rubber by means of sulphur or sulphur-containing compounds. However, by the use of certain organic peroxides, successful vulcanization is accomplished. Silicone-rubber stocks can be molded in conventional rubber molds using a mold cycle of from 3 to 10 min and a mold temperature of 125–150 C.

After the silicone-rubber stock has been fabricated into finished parts by either molding or extrusion, a bake cycle of from 50 to 60 hr at 200 C is necessary to bring out its optimum properties.

PHYSICAL PROPERTIES

A detailed examination of the properties of the rubber will show best the stability of this material, Table 1. In this table are listed the properties of four silicone-rubber compounds.

TABLE 1 PHYSICAL PROPERTIES OF SILICONE RUBBER

	12600	12601	12602	12603
Original Properties				
Hardness (Shore "A" Durometer)	45-55	55-65	65-75	75-85
Tensile Strength, psi	200	450	650	500
Elongation, %	200	150	110	100
Tear Resistance, lbs./in	35	45	45	35
After 500 Hours at 200°C (392°F)				
Hardness Change	(+)15-20	(+)15-20	(+)8-10	(+)6-8
Tensile, % Change	(+)20	(+)5	± 0	± 0
Elongation, % Change	(-)50	(-)50	(-)25	(-)25
Weight Loss, %	2-3	2-3	2-3	2-3
Low Temperature Resistance				
Flex test after 6 hrs. at (-) 60°C	No cracks or crazing	No cracks or crazing	No cracks or crazing	No cracks or crazing
(-) 76°F				
Compression Set				
Compression 30% 22 hrs. at 70°C (158°F)	10%	15%	10%	10%
Compression 30% 6 hrs at 150°C (302°F)	30%	40%	25%	25%

These compounds differ as to formulation and range in hardness from 45 to 85. The original mechanical properties are characterized by low tensile strength, moderate elongation, and low tear resistance. However, after 500 hr at 200 C heat-aging, the mechanical properties have undergone no great change. Flexibility is retained, tensile strength has suffered no change, and weight loss is low. At 200 C natural and synthetic rubbers decompose chemically after 10–15 hr. At –60 C the flexibility of silicone rubber is maintained with no cracking or crazing.

Compression-set tests on natural and synthetic rubbers are normally run at 70 C. At this temperature the compression set of silicone rubber averages 10 to 15 per cent. When the material is compressed 30 per cent and held for 6 hr at 150 C, compression-set values average 25 to 40 per cent, i.e., the rubber regains 60 to 75 per cent of the amount compressed.

While it can be easily understood that there would be many misgivings as to the strength of the material at room temperature, compare the tensile strength of silicone rubber with two synthetic varieties at the operating range of 200 C, as shown in Fig. 4. Both synthetic elastomers decreased in tensile strength 90 per cent within 60 min and both suffered chemical decomposition in 15 hr. The three silicone-rubber formulations retained their tensile strength and flexibility over a period of 800 hr in this particular test.

The stability of silicone-rubber formulations as to hardness and elasticity is shown by the following tests that were carried out by measuring with a Shore A durometer, the hardness over a range of temperatures from –60 C to 200 C, Fig. 5. The four silicone formulations showed less than 15 per cent change, while the curve of the two synthetic varieties run in comparison shows a change of 70 to 80 per cent.

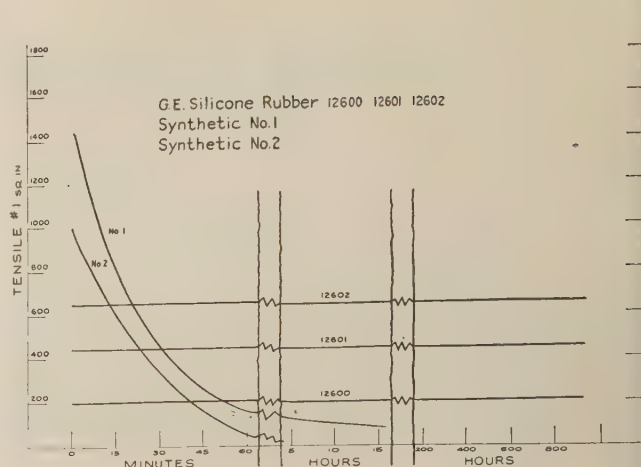


FIG. 4 CHANGE OF TENSILE STRENGTH OF SILICONE RUBBER AND TWO SYNTHETIC RUBBERS WHEN AGED AT 200 C

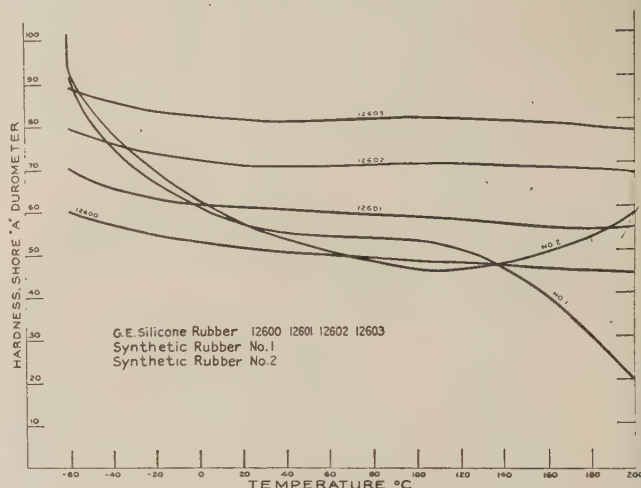


FIG. 5 CHANGE IN HARDNESS—SHORE A DUROMETER—WITH TEMPERATURE OF SILICONE RUBBER AND TWO SYNTHETIC VARIETIES

FIELD OF USEFULNESS

As to where the properties of silicone rubber may be put to use, the field of gaskets and packings immediately suggests itself. In the design of seals, engineers are looking with greater favor on molded elastomeric materials. The properties of silicone rubber which offer flexibility and long life at elevated temperatures make this material of great use in a variety of designs. Principal among these is the use of the material as a gasket to provide static seals where hot air under pressure is being transferred. The following designs have proved successful in this application:

In a 16-cylinder Diesel engine, the junction of each cylinder and supercharger connection was sealed with a 5-in.-diam molded silicone gasket of a rectangular cross section, Fig. 6. Operating conditions were 300 F and 15-psi air pressure. The heat-aging of the silicone rubber in this case eliminated frequent replacement of the gasket.

In early-model aircraft turbochargers, Fig. 7, a large O-ring gasket was necessary for proper sealing action. Operating conditions here were 400 F. A 3/16-in. extruded silicone-rubber rod bonded at the ends to form an O-ring provided successful gasketing action.

In a particular drying oven, Fig. 8, a large area had to be sealed between the water scrubber and the drying channel. Here again

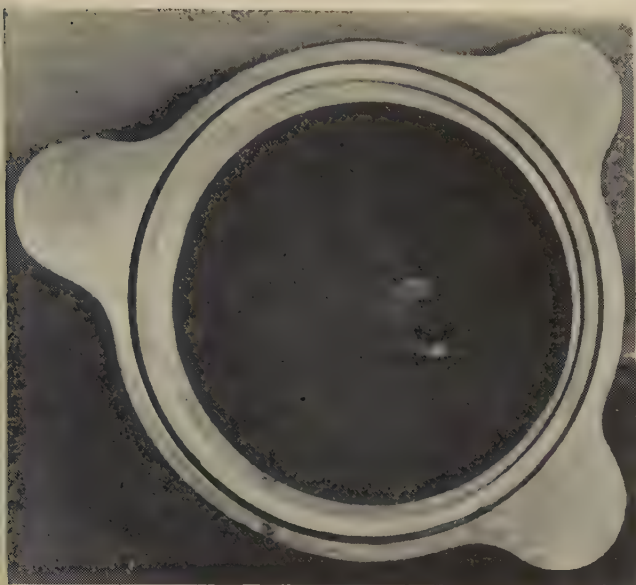


FIG. 6 SILICONE-RUBBER GASKET USED TO SEAL SUPERCHARGER-CYLINDER CONNECTION OF A DIESEL ENGINE



FIG. 7 SILICONE-RUBBER GASKET USED IN TURBOCHARGER

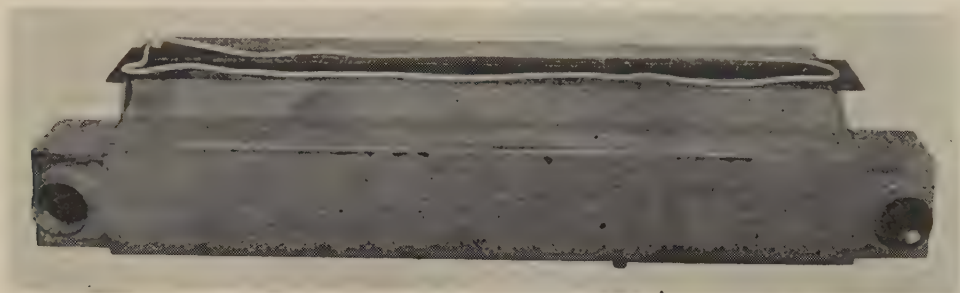


FIG. 8 SILICONE-RUBBER GASKET SEALING LARGE AREA BETWEEN TWO UNITS OF DRYING OVEN



FIG. 9 SILICONE-RUBBER V-PACKING FOR OIL SEAL

the temperature was above 300 F, and the air contained several solvents. Effective sealing action was obtained by using a large O-ring gasket with a $\frac{3}{8}$ -in. cross section.

O-ring packings are an excellent design when considering silicone rubber as a gasket material because of the fact that the packing friction on O-rings is very low. Then too, in the production of large-diameter O-rings, circular extrusions are simply bonded together, thus eliminating costly molds.

V-rings, or chevron packings, can be molded from silicone-

rubber stock and used to increase the operating temperature of oil seals, Fig. 9. Well-housed V-packings of silicone rubber have successfully provided a seal around a high-speed shaft where the oil temperature was 300 to 400 F. Shaft seals of another design have been produced by molding the packing inside of a metal-ring housing, Fig. 10.

In general, silicone rubber is not recommended for packing designs where a large surface area of the packing itself is exposed to a rotating shaft, nor is it recommended for packing in contact

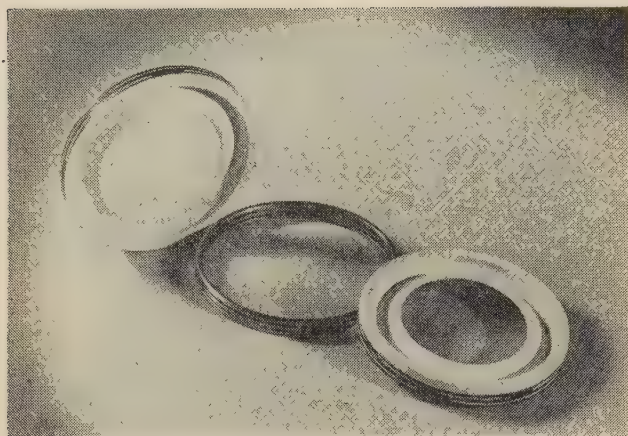


FIG. 10 SILICONE-RUBBER SHAFT SEALS PRODUCED BY MOLDING RUBBER INSIDE OF METAL RING

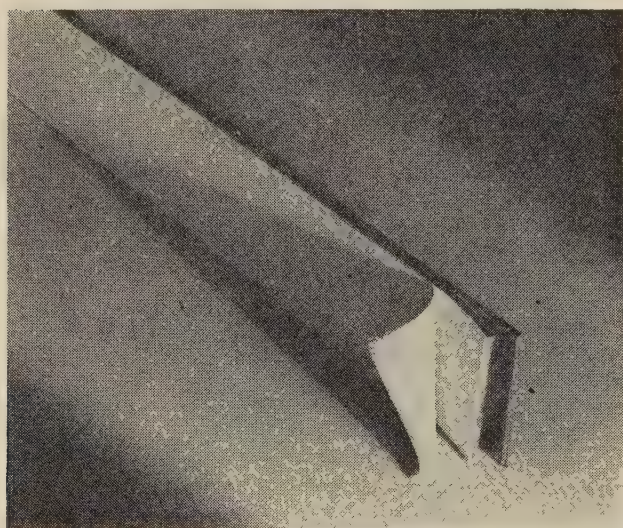


FIG. 11 SILICONE RUBBER BONDED TO METAL STRIP WITH GLASS-CLOTH FACING

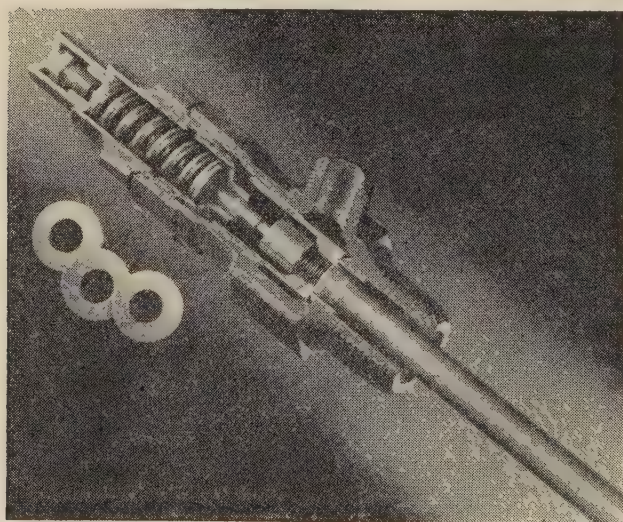


FIG. 12 SILICONE-RUBBER DISCHARGE-VALVE SEAT FOR ELECTRIC HOT-WATER HEATER

with steam. However, where seals are necessary in mechanical systems involving various lubricating oils or air at elevated temperatures, properly designed silicone-rubber gaskets and seals give exceptional service.

In the design, Fig. 11, silicone rubber is used to provide a flexible cushion for a metal strip that is to operate at an elevated temperature. An extruded flat section of rubber is bonded to the metal with a special adhesive. In such a case, if abrasion is a problem, silicone-coated glass cloth can be bonded to the outer surface of the rubber. Thus the desired property of flexibility after heat-aging is attained, and mechanical weakness minimized.

The surface of the fully cured rubber is entirely free of tack. There is no tendency for it to stick or adhere to metallic or non-metallic surfaces. This lack of adhesion extends over the entire operating range of the rubber which is not true for other elastomers. Advantage has been taken of this characteristic in packing designs, where heretofore adhesion has been a major problem, Fig. 12. In the discharge-valve seat of an electric hot-water heater, high temperatures and no adhesion were the problems. A silicone-rubber valve seat met both requirements satisfactorily.

CHEMICAL PROPERTIES

The exact effect of solvent action and chemical corrosion is difficult to determine by purely laboratory means. Immersion tests, Table 2, do not tell the entire story. Especially in the application of gaskets and seals, the surface area of exposure, the temperature at the point of exposure, the compressive force on the gasketing material all have a direct bearing on the degree of corrosion and swelling. In general, hydraulic fluids, lubricating oils, various polar solvents, dilute acids, alkalis, and concentrated salt solution cause very little swelling action on silicone rubber at room temperatures. Concentrated acids and alkalis have a very corrosive action on the rubber and nonpolar solvents impart high swelling.

In gasketing applications involving the use of hydraulic fluids, lubricating oils, and solvents at elevated temperatures, it is recommended that the silicone gasket have as little surface area exposed to the fluids as possible. An O-ring design, confined in a chamber under pressure to effect a seal, has worked well in most cases. Silicone-rubber packings with a square or rectangular cross section have also worked well.

An O-ring design of silicone rubber has provided an effective oil-and-air vapor seal on the TG-180 jet engine. The temperature at the point of seal was 350 to 400 F. No swelling was observed from the effects of the oil vapor and the gasket could be re-used.

The harder silicone-rubber stocks with silica as a filler are much more resistant to all chemical action and are recommended for use where corrosion and swelling are problems.

Steam has a very corrosive action on silicone rubber when the rubber is totally surrounded by steam. Tests have been run involving low-pressure steam (20 psi), and it was found that the silicone polymer was leached out of the rubber. A 5 to 10-mil surface area of the rubber samples lost 35 per cent of their polymer in 500 hr. However, silicone-rubber parts placed in metal housings with only a small surface area exposed suffered decomposition only at the point of contact with the steam. This opens the possibility that if elastomeric gaskets are needed for sealing at elevated temperatures where steam is present, a well-housed silicone gasket may well serve the job.

ELECTRICAL PROPERTIES

In Table 3 is listed a summary of the electrical properties of four grades of silicone rubber. From consideration of the di-

TABLE 2 CHEMICAL RESISTANCE OF SILICONE RUBBER

	12600		12601		12602		12603	
Reagent	% Change Wt.	% Change Volume	% Change Wt.	% Change Volume	% Change Wt.	% Change Volume	% Change Wt.	% Change Volume
Water	+2.4	+7	+0.6	+1	+0.6	0	+0.59	0
Water at 100°C	+7.1	+14	+6.8	+15	+3.2	0	+3.0	0
H ₂ SO ₄ conc.	dissolved		dissolved		dissolved		dissolved	
HCl conc.	+11.7	+42	+24.5	+43	+2.4	+3	+2.0	+8
HNO ₃ conc.	dissolved		+82.3	+118	+7.3	+8	+6.4	+8
NH ₄ OH conc.	+4.9	+13	+5.8	+11	+2.1	0	+2.7	0
KOH 50%	dissolved		dissolved		-3.0	+1	-4.5	-5
H ₂ SO ₄ 10%	+16.0	+37	+18.0	+38	+0.9	0	+0.5	0
NH ₄ OH 10%	+4.5	+13	+4.4	+10	+1.7	0	+1.7	0
NaCl conc.	+0.2	+8	+0.1	0	-0.1	-1	+0.1	+1
Mineral Oil at 25°C	+2.5	+2	+2.5	+3	+3.0	+4	+2.5	+2
Mineral Oil at 168°C	dissolved		---	+29	---	+38	---	+42
Castor Oil	+3.0	0	0	0	0	+1	+0.1	+1
Linseed Oil	+3.8	+3	+0.3	-1	-0.4	+3	+0.4	+2
Glycerine	+0.5	0	0	-1	+0.3	+1	0	0
Alcohol	---	+5	---	+5	---	+5	---	+5
Acetone	---	+25	---	+25	---	+25	---	+25
Benzene	---	+160	---	+150	---	+85	---	+85
Toluene	---	+200	---	+200	---	+85	---	+85

TABLE 3 ELECTRICAL PROPERTIES OF SILICONE RUBBER

	12600	12601	12602	12603
Dielectric Strength s/s	300	260	380	400
vpm at 25°C s/t	320	280	450	450
Dielectric Strength s/s	300	250	510	380
vpm at 100°C s/t	350	330	560	500
Dielectric Strength s/s	350	330	620	500
vpm at 150°C s/t	400	340	680	500
Power Factor, 1 Mc. dry	.0007	.005	.003	.004
3 days wet	.011	.017	.004	.005
7 days wet	.014	.018	.005	.005
14 days wet	.016	.018	.005	.005
Dielectric constant, 1 Mc. dry	3.8	7.4	3.1	3.1
3 days wet	4.4	8.6	3.2	3.2
7 days wet	4.5	8.6	3.2	3.2
14 days wet	4.9	8.9	3.2	3.3
Volume resistivity, megohms/cc	1.5x10 ⁵	1.5x10 ⁵	1.5x10 ⁵	1.5x10 ⁵

thetic varieties are compared over a temperature range of from room temperature to 200 C. The natural rubber remains fairly constant at 700 volts per mil, the two synthetic varieties decrease rapidly from 600 to 200 volts per mil, while the silicone rubber increases from 400 to 600 volts per mil.

When dielectric-constant and loss-factor values at 60 cycles are compared over a range of temperatures, the dielectric constant remains constant at 3.2, but the loss factor increases with increasing temperature from 0.01 at 25 C to 0.24 at 150 C, Fig. 14. Dielectric-constant and loss-factor values were determined over a range of frequencies, and, here again, the dielectric constant remained unchanged, while the loss factor rose rapidly after 1000 megacycles, Fig. 15.

In all electrical applications, ozone resistance is important. Silicone-rubber samples have stood up well when exposed to low concentrations of ozone.

ELECTRICAL APPLICATIONS

Advantage is taken of a combination of properties in the use of silicone rubber as capacitor bushings, Fig. 16. Good dielectric strength is of prime importance, coupled with the fact that the rubber, containing no plasticizer, does not contaminate the pyranol in the capacitor. The low-compression-set characteristics provide a permanent seal, and the design, in general, lends itself to assembly-line production.

The insulation properties and those of heat stability can also be put to use when silicone rubber is used as a wire-coating material. The coating of wire and cable with silicone rubber has been under investigation for some time with respect to formulation of the rubber and proper methods of fabrication. At the present time, silicone-coated wire can be produced and has been used successfully on motor-lead wires and electrical cable where temperature was a problem. Glass braid can be woven over the rubber-coated wire to improve the abrasion resistance of the rubber.

FABRICATION

The fabrication of finished parts from the compounded silicone stocks can be accomplished in many ways and this versatility simplifies greatly design problems. The most rapid fabrication method from a production standpoint is extrusion. By this

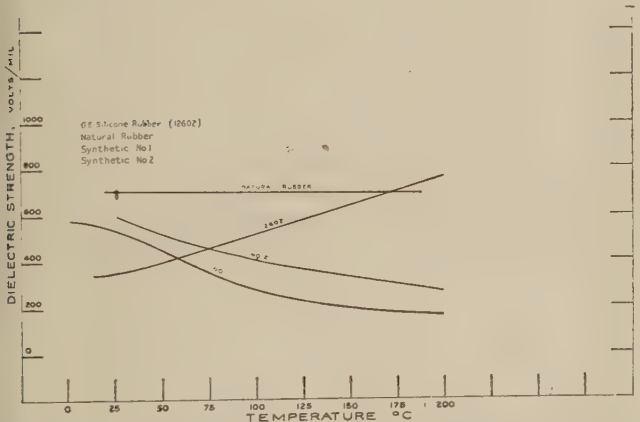


FIG. 13 DIELECTRIC STRENGTH VERSUS TEMPERATURE OF SILICONE RUBBER, NATURAL RUBBER, AND TWO SYNTHETIC VARIETIES

electric-constant and power-factor values, it can be seen that the harder stocks, with silica as a filler, have the best electrical properties. Of particular interest is the increasing dielectric-strength values with increasing temperature.

This is shown more graphically in Fig. 13 where the dielectric-strength values of silicone rubber, natural rubber, and two syn-

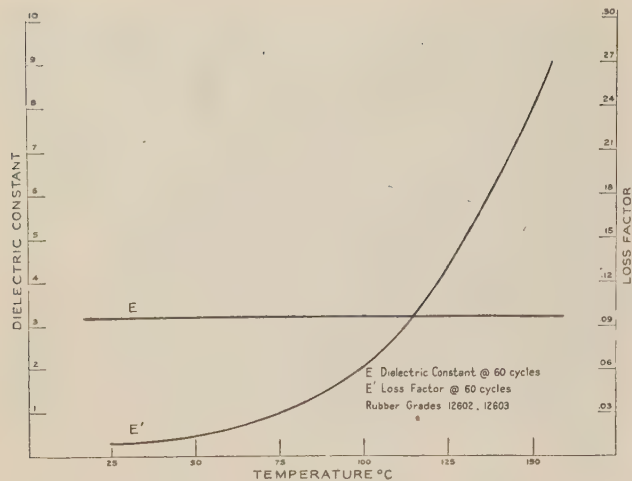


FIG. 14 DIELECTRIC CONSTANT AND LOSS FACTOR OF SILICONE RUBBER VERSUS TEMPERATURE

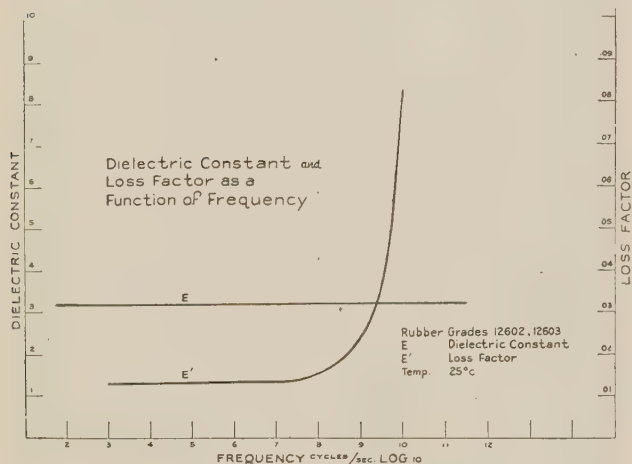


FIG. 15 DIELECTRIC CONSTANT AND LOSS FACTOR OF SILICONE RUBBER AS FUNCTION OF FREQUENCY

process a variety of shapes and sizes can be produced to tolerances of ± 0.005 in.

Molded parts can be produced in conventional rubber molds making use of compression, transfer, or injection-molding techniques. Optimum mold pressures are low, 100–500 psi, and optimum mold temperatures are from 125–150 C.

Coating of various cloths can be accomplished by use of conventional knife or dip-coating machines. When glass cloth is coated with silicone rubber, the resulting product is a flexible

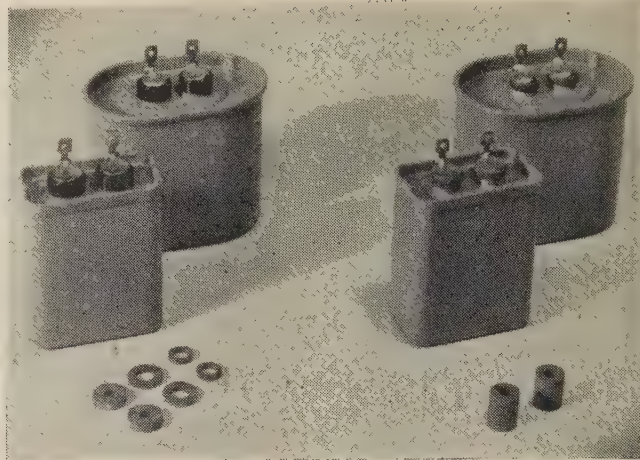


FIG. 16 SILICONE RUBBER USED AS CAPACITOR BUSHINGS

rubberized cloth that has been used as insulation-wrapping tape for electrical coils, and also as a thin gasketing material. Laminations or reinforced sheeting over $\frac{1}{8}$ in. thick can be produced by bonding together alternate layers of rubber and cloth. By this means tensile strengths of 2000–3000 psi and tear strengths of 200–300 lb per in. are obtained. This is highly advantageous when strength is needed, and elongation is not necessary. Bonding of the rubber to itself, glass or metal, can be accomplished with the use of a newly developed adhesive.

All these fabrication techniques or combinations of them may be utilized when parts of varied designs and structures are required.

CONCLUSION

In conclusion, it can be pointed out that silicone rubber is in no way a substitute material or just another elastomer. Its properties of thermal stability offer to the materials engineer a brand new product and a new range of temperature conditions where elasticity can be obtained.

Its flexibility range of from -70 F to 500 F, its resistance to oxidation, excellent heat-aging, low compression set, and good electrical properties are factors which can be put to use in a variety of applications. However, to take full advantage of these desired properties the limitations of the material should also be understood. These limitations can be overcome to a large extent by proper mechanical design. Fabrication techniques of molding, extrusion, wire-coating, laminating, cloth-coating, and bonding have been worked out successfully. The properties of silicone rubber combined with proper design and fabrication will result in products designed to solve many engineering temperature problems.

Development of a High-Speed Lathe for Machining Aluminum

By R. L. TEMPLIN,¹ NEW KENSINGTON, PA.

This paper describes an experimental high-speed lathe, having a maximum spindle speed of 9000 rpm, capable of taking a 2 $\frac{1}{4}$ -in.-diam bar through its hollow spindle, driven by a special 80-hp motor; together with certain accessory equipment. Certain machining experiments conducted with the lathe on two aluminum alloys at cutting speeds varying from 5000 to 20,000 fpm are described.

THE excellent machining qualities of aluminum have been known for many years, but the maximum speeds attained during the recent war period, particularly in machining aircraft parts, emphasized the possibility of even higher cutting speeds. The practical limitations on the maximum machining speeds for aluminum so far have been imposed by the limitations of the machine tools rather than by the machining properties of aluminum. Approximate maximum spindle speeds currently available in machine tools of the type indicated are shown in Table 1.

TABLE 1 TYPICAL MAXIMUM SPINDLE SPEEDS FOR CONVENTIONAL MACHINE TOOLS

Machine tool	Type	Size	Typical maximum spindle speed, rpm
Lathe	Engine	14-in.	1200
	Screw machine		5000
	Turret lathe	No. 2 $\frac{1}{2}$	3500
Milling machines	Plain or universal	No. 3	1500
	Spar millers		3600
	Router		6000
	Floor		1800
Drill presses	Sensitive	20-in.	10000
		$\frac{1}{4}$ -in.	

Milling-machine spindles have been operated experimentally at speeds ranging from 6000 to 9000 rpm, corresponding to cutting speeds of 5000 to about 20,000 fpm, using the high-cycle type of motor drive.² Under such conditions of operation the spindle bearings failed rather quickly because they were not designed for these conditions. As would be expected, it was found that higher spindle speeds required appreciably more horsepower than normally provided for the machine tools. It seems evident, therefore, if the advantages of the excellent machining properties of aluminum are to be realized, that the maximum speeds of machine tools must be increased considerably. This will require not only more power, but also consideration of many other factors which will be affected by the higher speeds.

Available cutting-tool materials of the cemented-carbide type appear to be capable of performing satisfactorily at any cutting speeds so far used for machining aluminum. If machine tools are

developed and made available for operating at much greater speeds, there is reason to believe that cutting-tool materials can be improved to function satisfactorily at these higher speeds. So far, however, there is little if any evidence to indicate that the currently available cutting-tool materials will not be adequate for the very high machining speeds when used on aluminum.

Because of its interest in extending the use of higher cutting speeds for machining operations on aluminum, the author's company, in 1944, decided to initiate the development of an experimental high-speed lathe capable of doing turning operations at speeds well beyond those currently available in conventional commercial lathes. This development has been pursued with the co-operation and assistance of leading manufacturers of machine tools, bearings, electrical equipment, and cutting-tool materials. Within the organization, the development was closely followed by a committee composed of representatives from the electrical, mechanical, and research departments, stimulated by a real interest on the part of top management.

THE EXPERIMENTAL LATHE

In the development of an experimental machine of this type, it was necessary to select the size of the machine to be designed and built. After consideration of the many factors involved, it was decided to develop an experimental turret-type lathe with a spindle having an internal bore of about 2 $\frac{1}{2}$ in., which would operate at a maximum speed between 7500 and 10,000 rpm. Two serious problems were immediately evident: What kind of bearings and what type of drive should be used for the machine? Consideration was given to ball, roller, and plain bearings before deciding to use the "Filmatic"³ segmental type. Although bearings of this type and of the size needed had never before been operated at the top speed specified, yet the manufacturer thought this type of bearing would function satisfactorily under the conditions indicated, with the result that these bearings were selected for both the radial and thrust bearings of the lathe spindle, and the radial bearings of the motor.

The Cincinnati Milling and Grinding Machines, Inc., co-operated in the design and making of the spindle and motor bearings, and the spindle and motor housings. This company also made the motor shaft and assembled the special motor rotor on it. A longitudinal section of the spindle is shown in Fig. 2. The motor bearings are lubricated with oil from a central system at 4 psi pressure. The motor-bearing shoes are of the fixed type and are arranged with 0.005-in. diametral clearance on the shaft. The bearing shoes for the spindle are adjusted by hydraulic pressure which is regulated by a manually operated control valve. An oil pressure of 18 psi is maintained in the bearing chamber of the spindle housing for lubrication of the shoes.

It was recognized initially that suitable provision would have to be made for controlling the temperature of the lubricating oil used for the spindle and motor bearings. Therefore a 5-hp hermetically sealed cooling unit was obtained from the Chrysler Air-Temp Corporation and used to control the temperature of both the lubricating oil and the hydraulic oil used for operating the

¹ Assistant Director of Research and Chief Engineer of Tests, Aluminum Company of America, Aluminum Research Laboratories. Mem. ASME.

² "High-Speed Milling in Aircraft Production," *The Iron Age*, vol. 152, Oct. 14, 1943, p. 74, and "Machining and Forming 14S-T Aluminum Alloy," *The Iron Age*, vol. 152, Nov. 11, 1943, p. 65.

Contributed by the Machine Design Division and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-41.

³ "The Wedge-Shaped Oil Film Does the Trick," by John Haydock, *American Machinist*, vol. 83, 1939, p. 823.

feeding mechanism. A centrifugal pump was attached to the oil-storage-tank suction line to circulate the oil through the "Air-temp" unit and return it to the tank.

Various types of drive, including steel belt, V-belt, roller chain, gear, and direct connected, were considered initially, with the final choice being the last named. The selection of this type of drive increased the complexity of the electrical equipment considerably but did provide a workable solution to the immediate problem presented. The high speed of the spindle with a direct-connected motor indicated a somewhat conventional squirrel-cage-type motor which was adopted. The motor was designed with a hollow shaft, having an internal bore corresponding to that of the lathe spindle.

The spindle motor, which was furnished by the Reliance Electric and Engineering Company has a rating of 80 hp at 220 volts, 3 phase, 60 cycle, 3450 rpm. This same horsepower rating is maintained as the frequency is increased to 150 cycles, which corresponds to a top speed of about 8800 rpm. During this frequency and speed increase the voltage is raised to a maximum value of 350 volts. In order to handle the additional high losses resulting from starting and stopping, the stator is provided with water-cooling.

Since the project was entirely experimental, a power supply of one or more fixed frequencies was not considered suitable, as large gaps in the test data would result. Because of this, a rather elaborate electrical setup was provided, involving, in addition to the spindle motor, two motor generator sets. One of these sets is a conventional, constant-speed, a-c, d-c set, which provides a variable direct-current voltage from zero to 250 volts. This variable voltage supplies the motor of a second set comprised of a direct-current motor and alternating-current alternator which can be operated at any speed between zero and 1800 rpm, so that the lathe spindle can be operated at any speed desired between zero and 8800 rpm.

There were three main problems concerned with the control of this equipment. One of these was to provide a voltage input to the spindle motor proportional to speed from zero to 3450 rpm and then vary the voltage on a straight-line basis from 220 volts at 3450 rpm to 350 volts at top speed. Amplidyne control provided this voltage control and incorporated voltage forcing during periods of heavy demand on the alternator, such as exists during the accelerating period of the spindle motor.

The other control problems involved the provision of relatively quick starting and stopping of the spindle motor. During such starting and stopping, the speed of the alternator motor generator set is not varied, and when the motor is started it will come up quickly to whatever speed is represented by the preset speed of the alternator set.

In starting, for ultimate speeds above 3450 rpm, the spindle motor is thrown across the 220-volt 3-phase 60-cycle line in the conventional manner, and left there until it attains a speed of slightly below 3450 rpm. At this point the motor is transferred to the alternator and, simultaneously, the field of the alternator is forced during the remainder of the acceleration period. The motor then continues to accelerate to whatever speed corresponds to the preset frequency of the alternator set.

Assuming that the motor is operating at some speed above 3450 rpm, the stopping sequence consists in disconnecting the motor from the alternator and connecting it to the 220-volt 3-phase 60-cycle line, the rotation being such that regeneration (not plugging) occurs. This quickly slows down the motor, and when a speed slightly above 3750 rpm is reached, the motor is disconnected from the 60-cycle line, and direct current of approximately full-load value is sent through two legs of the stator winding. When the motor comes to a stop, this direct current is disconnected.

An alternative method of stopping employs plugging from the 60-cycle line for the deceleration below 3600 rpm. This has been tested with excellent results, but requires the additional complication of a zero-speed switch; therefore most of the operation has been done with the direct-current method of deceleration and stopping. A stop from full speed requires between 4 and 5 sec, and a similar time is required for acceleration.

The operator's control consists basically of a start push button, a stop push button, and a rheostat, which may be preset for any operating speed; and of course this speed may be changed while a cut is being made. The controls were furnished by The Electric Controller and Manufacturing Company.

It should be pointed out that most commercial applications of such a high-speed machine tool would require only a comparatively simple electrical setup, providing for one or two fixed frequencies. Such a scheme would be quite economical and might provide, for instance, for two operating speeds, one of 3600 rpm and one of, let us say, 7200 rpm. Quite confidently it may be expected that where a complete range of variable speeds is required, the electrical industry will within the next few years be able to provide an electronic frequency changer as a power source, which will eliminate a great deal of the expensive and bulky equipment used for this experimental installation.

The motor was connected to the back end of the spindle by means of a dynamically balanced, forged, aluminum-alloy, flexible coupling furnished by the Thomas Flexible Coupling Company. The ends of the coupling were pushed on the tapered ($1\frac{1}{2}$ in. per ft of diameter) ends of the spindle and motor shafts and retained by spanner nuts. No keys were provided and the friction of the tapered joints has proved quite satisfactory under the maximum torque loads used.

A hydraulically operated tool carriage with a separate hydraulic circuit and Vickers pump for operating the carriage longitudinally was provided by the author's company. A manually controlled operating valve, mounted on the side of the spindle housing, controls the carriage movement through a suitable hydraulic cylinder. Vickers cam-operated shutoff valves were provided in the same circuit for stopping the carriage at any desired point in either direction, within its maximum range. A manually operated cross-slide, and a turret-type toolholder were mounted on the tool carriage.

Advance feed speed of the tool carriage is controlled by a Vickers flow-control valve mounted on the front of the machine bed, and a rapid-return speed is obtained by allowing full-line oil volume to pass to the cylinder when the operating valve is set for return. A maximum feed of 120 ipm was provided. Following the first tests of the lathe, changes were made in the tool carriage and lathe bed so that the carriage could be fed transversely by the hydraulic mechanism. This arrangement was effected for the purpose of carrying out high-speed milling tests in which the cutter was mounted on the spindle nose and the work clamped on the tool carriage.

A heavy welded-steel bed was provided for the machine by the author's company, on which the motor, spindle housing, and tool-carriage assembly were mounted. The interior of the machine bed is used for an oil-storage tank which is provided with suitable baffles and filters. The hydraulic pumps and pressure-control valves are mounted on the rear plate of the bed. The machine requires about 150 gal of oil for the combined hydraulic and lubricating systems.

A collet chuck of $2\frac{1}{4}$ in. maximum capacity was supplied by the Warner and Swasey Company. This chuck is operated manually by a lever mounted on top of the spindle housing. In addition, a special 10-in. 3-jaw scroll-type chuck was furnished by the same firm. The body of this chuck is made from an aluminum-alloy 14S-T forging. Provision was made for bolting this chuck to the

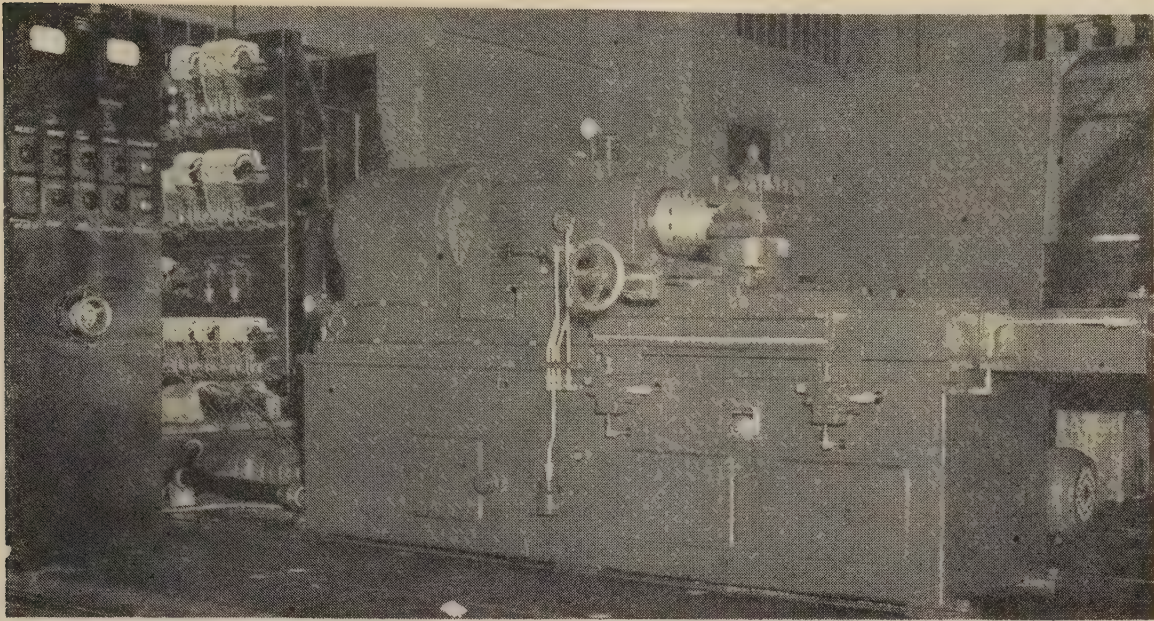


FIG. 1 EXPERIMENTAL HIGH-SPEED LATHE AND CONTROL PANEL

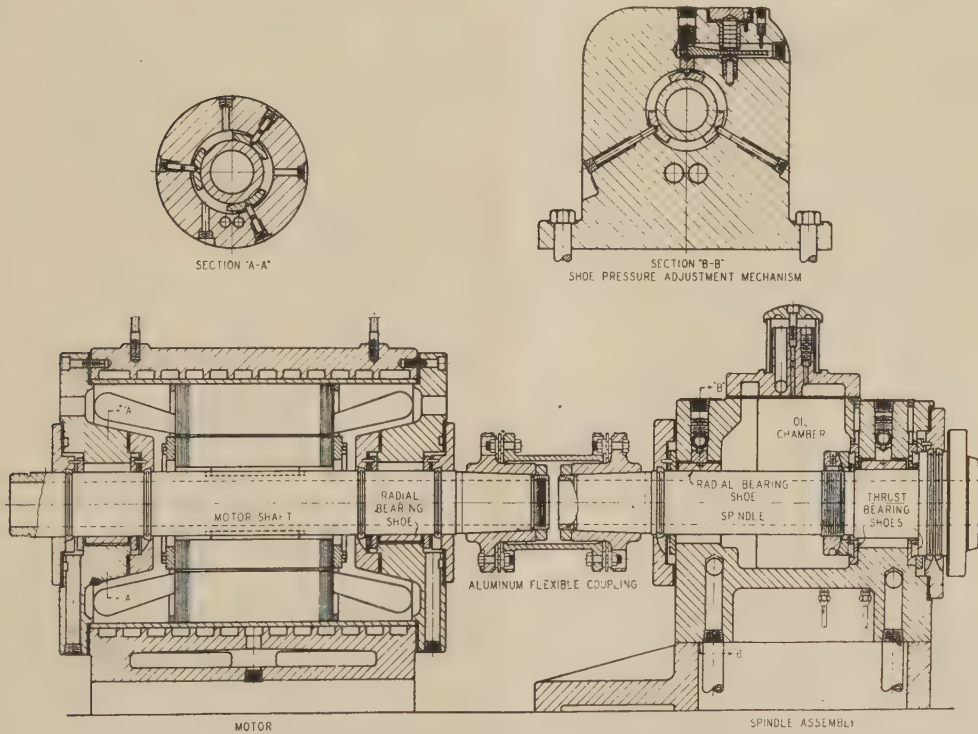


FIG. 2 SECTION OF EXPERIMENTAL HIGH-SPEED-LATHE SPINDLE ASSEMBLY

spindle nose of the lathe, and the chuck jaws are adjusted by the usual type of wrench.

The assembled lathe is shown in Figs. 1 and 2. The hydraulic diagram is shown in Fig. 3, and the wiring diagram for the equipment is shown in Fig. 4.

MACHINING EXPERIMENTS

The preliminary machining tests were conducted on 8-in-diam Alcoa 14S forging stock in the as-rolled and fully heat-treated (-T) tempers and on 13-in-diam disks cut from 3 and 4-in-thick

24S-T plate. The stock had the mechanical properties shown in Table 2.

TABLE 2 MECHANICAL PROPERTIES OF STOCK USED				
Alloy and temper	Tensile strength, psi	Yield strength, psi	Elongation in 2-in., per cent	Bhn
14S as-rolled.....	33600	14350	15.5	58
14S-T.....	68650	60100	6.5	130
24S-T.....	60200	47400	4.5	118

All of these tests were performed employing a single-point lathe

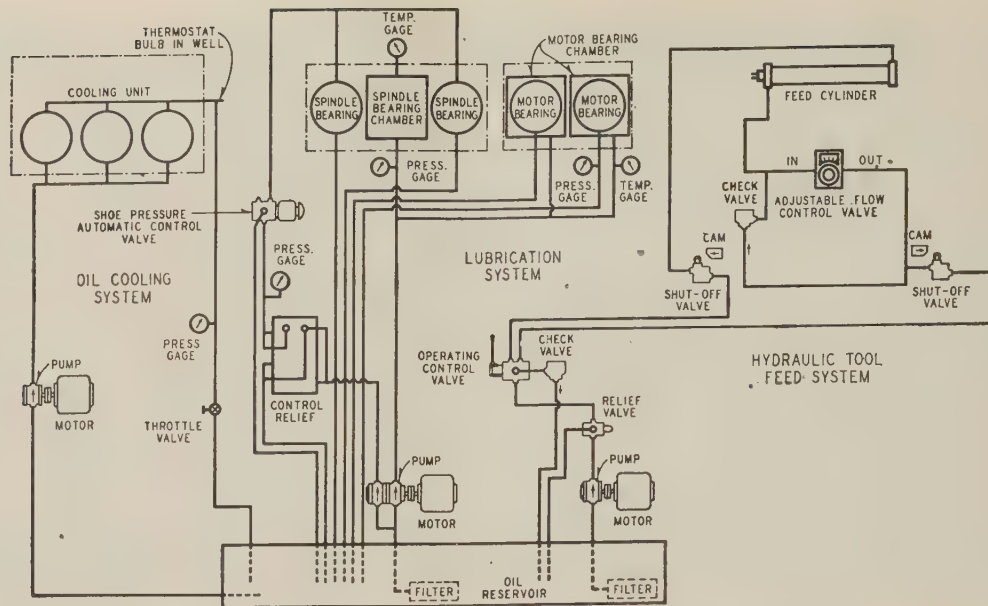


FIG. 3 DIAGRAM OF HYDRAULIC SYSTEMS FOR OIL COOLING, LUBRICATION, AND TOOL-CARRIAGE FEED

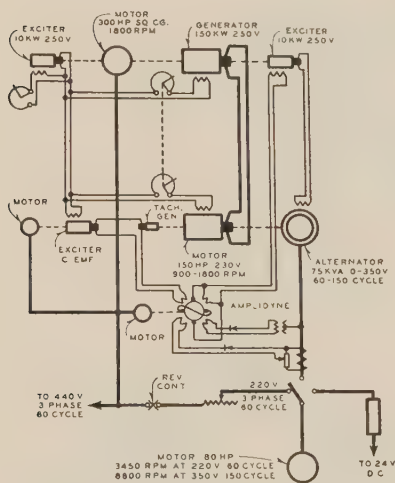


FIG. 4 SIMPLIFIED WIRING DIAGRAM FOR ELECTRICAL EQUIPMENT AND CONTROLS

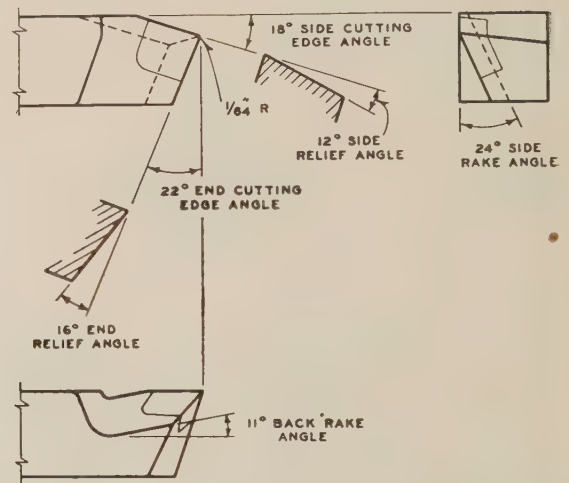


FIG. 5 SINGLE-POINT CARBOBOY NO. 44A LATHE TOOL USED IN CUTTING TESTS

tool tipped with Carboboy No. 44A material, as shown in Fig. 5. Other similar tools were employed in tests to improve the quality of the machined surface.

The 8-in.-diam stock was machined into flanged workpieces in conventional machine tools. These workpieces had a test section $5\frac{3}{4}$ in. diam $\times$ $9\frac{1}{2}$ in. long, and one is shown in the lathe in Fig. 1, and in detail in Fig. 6, item A. The details of the 13-in.-diam 24S-T disks are shown as item *b* in Fig. 6. In all of these tests the workpieces were bolted directly to the spindle nose by means of socket-head cap screws.

Because of the multitude of variables to be investigated and the fact that the carriage feed rate was independent of the spindle speed, a nomograph or alignment chart was developed to assist in adjusting the machine. This chart is presented in Fig. 7.

As an example of the use of this nomograph, assume a workpiece of 5.75 in. diam is to be machined at a cutting speed of 10,000 fpm, employing a 0.25-in. depth of cut and a feed of 0.018 in. per revolution. It is desired to find the proper speed of the lathe and the amount of metal which will be removed under these

conditions. By laying a straightedge through $D' = 5.75$ and $S' = 10,000$, we find that the required spindle speed at N' is 6700 rpm. Laying the straightedge through N' and $f' = 0.018$ we find $F' = 120$, which is the feed in inches per minute. Connecting $t' = 0.25$ and $D_m' = 5.5$ locates $A' = 4.3$ sq in., which is the area of the cut. Then connecting A' with F' we locate $V' = 520$ cu in. per min, which is the metal-removal rate under the conditions specified.

Cutting tests were conducted on the 8-in.-diam stock under the nominal conditions presented in Table 3, and on the 13-in.-diam disk under the nominal conditions presented in Table 4, so that the same volume of metal per minute would be removed under varying conditions of speed, depth of cut, and feed.

Additional cutting tests beyond the scope of these tabulations were conducted with feed rates as small as 0.00003 in., and as large as 0.025 in. per revolution. The maximum metal removal reported in these tests was 470 cu in. per min, or approximately 47 lb per min. This does not represent the maximum capabilities of the lathe.

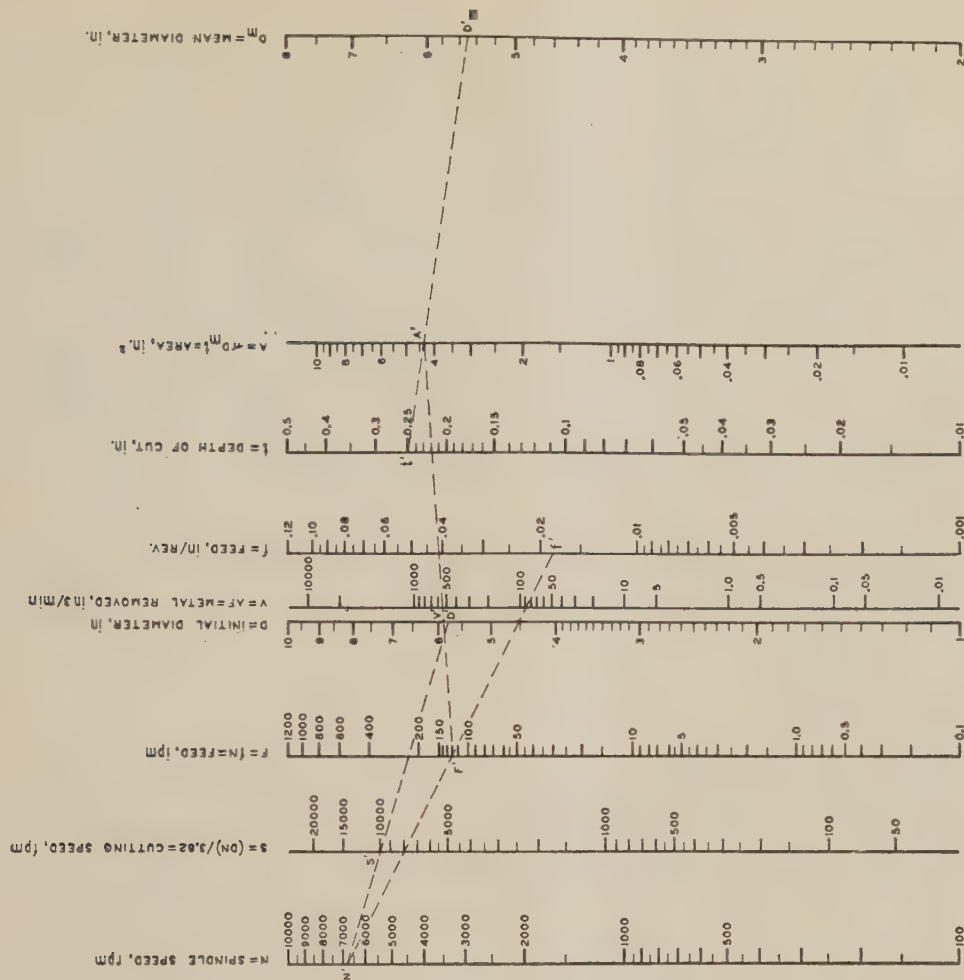


FIG. 7 ALIGNMENT CHART FOR EMPIRICAL DATA

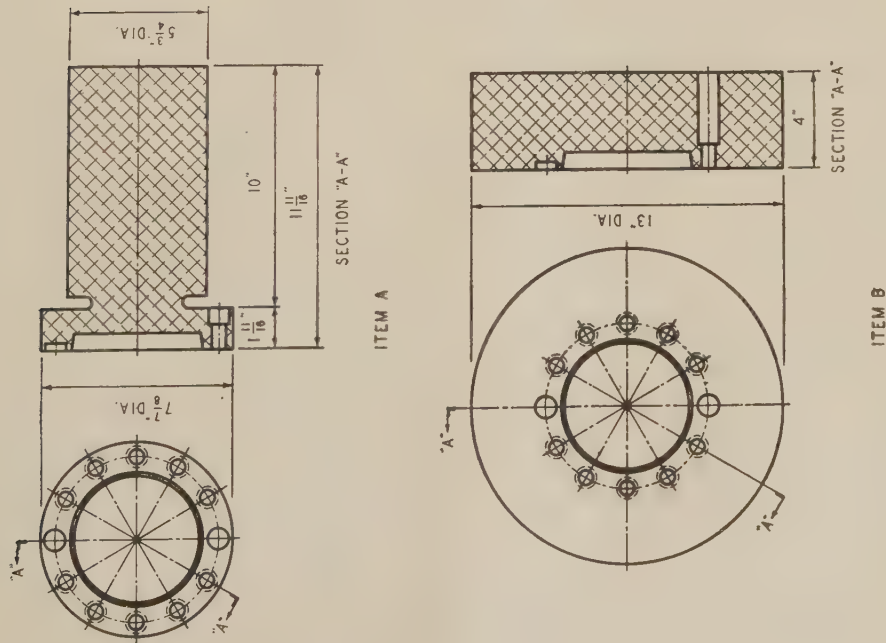


FIG. 6 ALUMINUM-ALLOY WORKPIECES FOR CUTTING TESTS

TABLE 3 VOLUME OF 14S METAL REMOVED, CU IN. PER MIN

Cutting speed, fpm	Depth of cut, in.	Nominal feed, in. per revolution			
		0.005	0.007	0.010	0.014
5000	1/8	35	49.5	70	99
7070	1/8	49.5	70	99	140
10000	1/8	70	99	140	198
5000	1/4	70	99	140	198
7070	1/4	99	140	198	280
10000	1/4	140	198	280	396

TABLE 4 VOLUME OF 24S-T METAL REMOVED, CU IN. PER MIN

Cutting speed, fpm	Depth of cut, in.	Nominal feed, in. per revolution				
		0.005	0.007	0.010	0.014	0.020
10000	1/8	75	106	150	212	300
14140	1/8	106	150	212	300	424
20000	1/8	150	212	300	424	600

In all tests the machine was set to nominal values of speed, feed, and depth of cut, but actual values were determined by measurement. Idle power, spindle temperature, and workpiece dimensions were measured before and after each cut. Total input power was measured during each cut. In many cases the dimensions and description of the chips were obtained. All cutting tests were performed dry without the use of any coolant or lubricant, and the tool was hand-stoned between cuts.

In the early tests it was found that a relatively large amount of power was required to drive the spindle even when a cut was not being made. Tests were conducted in an effort to reduce the additional power requirements in the following ways:

- 1 Varying hydraulic pressure on cylinders which control radial force on spindle bearings.
- 2 Increasing clearance on spindle thrust bearings.
- 3 Removing several shoes from spindle thrust bearings to provide less rubbing surface and a less obstructed flow of lubricating oil around the bearings.
- 4 Providing more clearance on spindle radial bearings.
- 5 Substituting a lighter oil in the lubrication system.

Fig. 8 is a log-log plot of idle horsepower versus spindle speed in revolutions per minute. Extrapolation of the curves in Fig. 8 shows that the 80-hp motor is loaded to about 70 per cent of its capacity under idle conditions at a spindle speed of 9000 rpm when the temperature of the heavier oil is in the range from 90 to 110 F. Permitting the oil temperature to rise to approximately 140 F resulted in an appreciable reduction in the idle horsepower for a given spindle speed. The spread in values is shown in Fig. 8. All power values presented represent motor output power converted from measured input power by means of data obtained from motor-efficiency tests conducted by the Reliance Electric and Engineering Company.

Increasing the clearance on the thrust and radial bearings and modifying the construction of the thrust bearings had little significant effect on the idle-power requirements. Some experimental work was done in which the hydraulic pressure in the radial-bearing shoe cylinders was varied. It was found that varying this pressure on the bearing shoes resulted in some change in the idle horsepower, providing this pressure was in the range of 0 to 150 psi. As shown in Table 5, changing the oil to a lighter grade resulted in an average decrease of about 17 per cent in the idle horsepower for the spindle. The data points for these tests are shown in Fig. 8 and they lie along the curve representing the maximum oil temperatures developed when the heavier oil was employed.

The efficiency of the cutting operation from a power viewpoint was determined by subtracting the idle horsepower from the total horsepower, resulting in the net horsepower for each cut. The number of cubic inches of metal removed per minute for each net horsepower is plotted against tool feed in inches per revolution in Fig. 9. The data points represent cuts in which the depth of cut

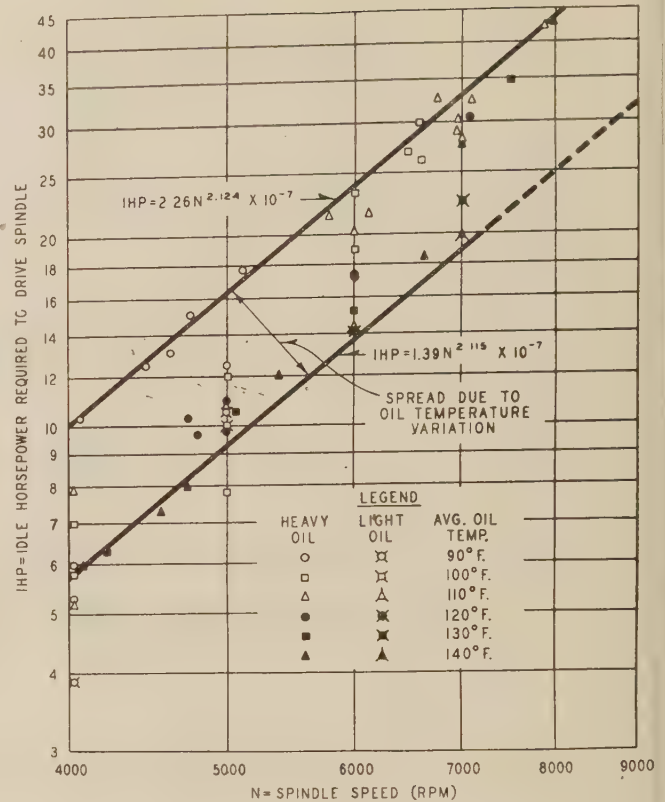


FIG. 8 VARIATION OF IDLE HORSEPOWER WITH SPINDLE SPEED

TABLE 5 EFFECT OF OIL VISCOSITY ON IDLE HORSEPOWER

Speed, rpm	Average spindle temp, deg F	Average idle horsepower		Decrease in average idle horsepower per cent
		Heavy oil	Light oil	
4000	90	4.65	4.3	7.5
5000	90	12.50	10.5	16.0
5000	100	12.40	10.0	19.3
6000	110	18.00	14.3	20.5
6000	120	17.10	14.1	17.5
7000	140	25.80	20.4	20.9
Avg				17.0

	Viscosity SSU/100 F	Interpolated viscosity SSU/140 F	Viscosity SSU/210 F
Heavy oil.....	107.0	58.9	39.1
Light oil.....	63.4	45.0	35.0

was either 1/8 or 1/4 in., and the surface cutting speed was approximately 5000, 7070, 10,000, 14,140 or 20,000 fpm. It is evident that the cutting efficiency from a power viewpoint is dependent largely upon the alloy and temper and the tool feed. The number of cubic inches of metal removed per minute per horsepower increases rapidly with increases in feed, and in these tests on 14S, a maximum of 7.5 cu in. per min per hp were removed for a feed of approximately 0.025 in. per revolution. Fewer cubic inches of metal per minute are removed for each horsepower when 14S-T or 24S-T is machined. When 24S-T was machined at a cutting speed of 20,000 fpm, the 80-hp motor was considerably overloaded in several instances (to 155 per cent of its rated capacity in the most extreme case) without any indication of stalling.

From the data presented in Fig. 9 calculations were made to determine the tangential tool load per inch of cut depth. This information is plotted in Fig. 10 against feed in inches per revolution. The calculated load on the tool increases with increasing rates of feed and is greater for the 24S-T than for 14S-T or 14S as-rolled material. The load on the tool per inch of cut depth appears to be quite independent of cutting speed in the range 5000

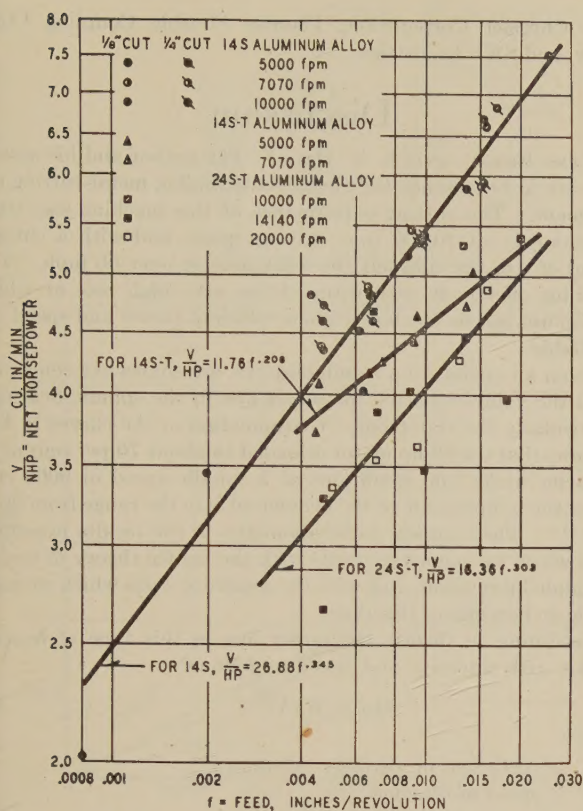


FIG. 9 RELATION BETWEEN VOLUME OF METAL REMOVED PER HORSEPOWER AND FEED

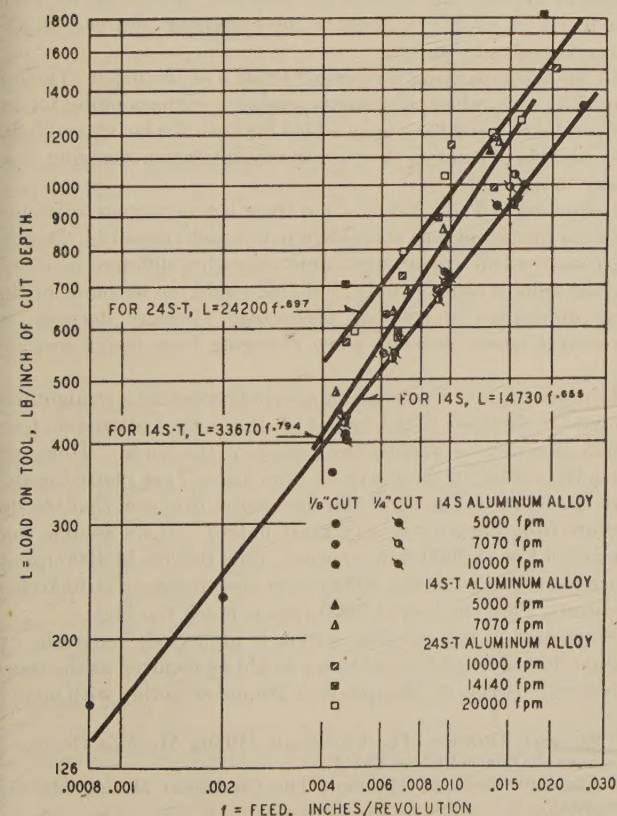


FIG. 10 RELATION OF UNIT TOOL LOAD TO FEED

to 20,000 fpm, as well as the actual depth of cut (for the range investigated).

In all of the tests, machining 14S-T resulted in relatively short broken chips, but continuous straight cuttings were obtained when as-rolled 14S was machined, providing the tool feed did not exceed approximately 0.010 in. per revolution. Beyond this value the chips began to curl in closer spirals as the feed was increased, and a feed was reached at which broken chips were obtained from the as-rolled 14S material. The width of the chips agreed very well with values calculated from the depth of cut and tool angles, but the over-all thickness of the chips ranged from 1.63 to 3.45 times the calculated values. This ratio was greater the smaller the rate of tool feed. Machining 24S-T at the higher cutting speeds between 10,000 and 20,000 fpm resulted in very short broken chips regardless of tool feed.

The ability of this machine to remove metal at such fast rates focuses attention on the problems generally associated with chip disposal. The relatively large kinetic energy imparted to the chips by the high cutting speeds can be utilized in their disposal. When the chips are short and broken, suitable shields or guards can and should be used to deflect the chips into suitable receivers or conveyers. When the cuttings are long and continuous, the tools can be designed to direct them away from the work and into a pile or container some convenient distance from the lathe.

No difficulties were encountered from chattering or vibration, and the diameter of the workpieces was quite constant throughout the length of cut. Better finishes were obtained for the lighter feeds, and the work remained relatively cool even when the metal removal was high, in spite of the fact that no cutting compound was employed. It was found extremely important to prevent the chips from contacting the finished work in order to maintain the desired machined-surface quality. It was evident that most of the heat generated at the cut was carried away in the chips, and the chips attained higher temperatures when they were curled or broken.

Some experimental work was done to determine the approximate degree of unbalance which would be permissible in a workpiece to be turned in this lathe. The test piece was a 13-in.-diam disk of 3-in.-thick 24S-T rolled plate. Unbalance was obtained by drilling a hole in the test piece a distance of 4.75-in. from the center, and various degrees of unbalance were obtained by varying the diameter and depth of the hole. The variations were such that each succeeding step approximately doubled the unbalance expressed in centrifugal force and are presented in Table 6.

TABLE 6 CONDITIONS OF TESTS FOR PERMISSIBLE UNBALANCE

Diameter of hole, in.	Nominal depth of hole, in.	Centrifugal force at 7000 rpm, lb
0.125	3/4	5.96
0.125	1 1/2	12.08
0.177	1 1/2	24.59
0.250	1 1/2	48.60
0.250	2 11/32 (Through)	96.00
0.358	2 11/32	197.00
0.500	2 11/32	382.00
0.719	2 11/32	794.00

Vibration was insignificant under the first condition but increased steadily as the degree of unbalance was increased. Under the last condition cited the vibration was quite pronounced at all points on the machine and particularly on the tool post and tool carriage. It was apparent from the tests that an unbalanced condition which produced a centrifugal force of more than about 700 lb would not be permissible with this machine.

The collet chuck and the special aluminum-alloy three-jaw scroll-type chuck were both tested with satisfactory results, although the collet chuck appears to be the better type for work at the high speeds employed.

CONCLUSIONS

From the results of these tests, conducted on the experimental high-speed lathe, in which as-rolled 14S and 14S-T stock and 24S-T plate were machined dry at surface cutting speeds up to 20,000 fpm, the following conclusions appear to be warranted:

1 Aluminum alloys can be machined satisfactorily by turning at cutting speeds up to 20,000 fpm. No indication of an upper speed limit was observed.

2 The experimental high-speed lathe performed very well in these tests, except perhaps for the relatively large amount of power required to drive the spindle when no cut was being made.

3 The number of cubic inches of metal which can be removed per minute for each horsepower going into the cut is greater for as-rolled 14S than for 14S-T or 24S-T, and increases with increasing rate of tool feed. In these tests as much as 7.5 cu in. of metal were removed for each horsepower developed at the cut.

4 The net horsepower required to remove a given volume of metal is independent of cutting speed and depth of cut.

5 As much as 470 cu in. of metal per min (approximately 47 lb) were removed without overtaxing the equipment. There was no indication that greater amounts could not be removed.

6 The amount of power required to drive the spindle when a cut was not being made was greater than had been anticipated. Extrapolation of the data indicate that the 80-hp motor might be operating at as much as 70 per cent of capacity when driving the idle spindle at 9000 rpm.

7 Increasing the clearance on thrust and radial bearings had little effect on the idle horsepower but varying the hydraulic pressure to the spindle-bearing-shoe cylinders did cause some variation in this idle power.

8 Changing the oil in the lubricating and hydraulic system to a lighter oil resulted in an average reduction in idle horsepower of approximately 17 per cent.

9 Satisfactory machined-surface quality can be maintained only if the tools are designed to direct the chips away from the work and prevent them from marring the finished surface.

10 Even though the cuts were performed without a cutting compound, the work remained cool and the heat was carried away by the chips.

11 In spite of the fact that more efficient turning operations from a power viewpoint can be achieved by employing large rates of tool feed, better finish and dimensional control can be obtained with smaller feeds. The feed of course should not be reduced below one which will assure proper cutting.

12 Equations have been presented to express mathematically the effects of the operating characteristics of the experimental high-speed lathe as well as the machining characteristics of as-rolled 14S, 14S-T, and 24S-T.

13 Efficient handling of the cuttings at the high machining speeds investigated requires some consideration but apparently the problems presented can be solved satisfactorily.

14 The maximum condition of unbalance which is permissible in turning with this lathe is that which would produce a centrifugal force of about 700 lb.

15 The collet chuck and three-jaw scroll-type chuck both performed satisfactorily at moderate speeds, but the collet chuck appears to be more suited for the higher speeds.

ACKNOWLEDGMENT

The author is indebted to the following companies who cooperated in the foregoing experiments: Cincinnati Milling and Grinding Machines, Inc., The Warner and Swasey Company, the Reliance Electric and Engineering Company, the Carboloy Company, Inc., The Electric Controller and Manufacturing Company,

The Chrysler Corporation, Thomas Flexible Coupling Company, and SKF Industries.

Discussion

HANS ERNST⁴ AND A. H. DALL⁵ The author and his associates are to be commended for this outstanding metal-cutting experiment. The cutting performance of this machine was truly sensational; at 10,000 fpm cutting speed, and with a cutting ratio⁶ of 0.6, the chip left the workpiece at over 60 mph. The machine effectively demonstrated the very high rate at which aluminum can be machined when sufficient power and speed are available.

From an engineering standpoint we are somewhat concerned with the values reported for power loss in the spindle bearings, particularly the statement, "Extrapolation of the curves in Fig. 8 shows that the 80-hp motor is loaded to about 70 per cent of its capacity under idle conditions at a spindle speed of 9000 rpm when the temperature of the heavier oil is in the range from 90 to 110 F." There appear to be anomalies in the results presented here which we cannot reconcile with the known theory of hydrodynamic lubrication, and with the results of tests which we have made on bearings of this type.

According to theory, the power loss in this type of bearing varies with viscosity and speed as follows

$$Hp = K\mu N^2 \dots \dots \dots [1]$$

where

μ = coefficient of viscosity (absolute)

N = speed of rotation

K = constant of bearing

Therefore in Table 5 of the paper, if we assume that all of the power is dissipated in the spindle bearings, the percentage decrease in average idle horsepower with the lighter oil, for each spindle speed, should be equal to the percentage decrease in absolute viscosity of the oil.

In the accompanying revision of Table 5 of the paper (Table 7 herewith), the values of absolute viscosity corresponding to the temperatures listed have been added for both the heavy and light oils; also the percentage decrease in viscosity when changing from heavy to light oil.

Inspection of Table 7 shows that there is a very great difference between the percentage changes in power and viscosity. Doubtless the actual oil temperatures were somewhat different from the average spindle temperatures, but this would not account for the large differences in ratio of percentage viscosity decrease to percentage power decrease when changing from heavy to light oil.

Equation [1] of this discussion also indicates that a straight line should be obtained if the values of Hp/μ were plotted on Cartesian co-ordinates against the square of the speed. However, when the values for the heavy oil from Table 7 are plotted in this way, as shown in Fig. 11 of this discussion, it is seen that the departure from linearity is very great indeed. If we assume that the power loss at 7000 rpm is correct, then the loss at 4000 rpm is much too low; conversely, if the power loss, measured at 4000 rpm, is correct, then the loss at 7000 rpm is much too high.

Since part of the rotating system is immersed, it may be expected that some additional losses would be incurred as the speed increases. However, the spindle is ground smoothly with no pro-

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⁵ Assistant Research Director, The Cincinnati Milling Machine Company.

⁶ Cutting ratio = normal feed increment divided by chip thickness.

TABLE 7 EFFECT OF OIL VISCOSITY ON IDLE HORSEPOWER^a

Speed, rpm	Temp, deg F	Heavy oil —		Light oil —		Decrease in hp, per cent A	Decrease in absolute viscosity, per cent B		Ratio B/A
		Idle hp	Absolute viscosity	Idle hp	Absolute viscosity				
4000	90	4.65	21.6	4.3	11.2	7.5	48		6.40
5000	90	12.50	21.6	10.5	11.2	16.0	48		3.00
5000	100	12.40	19.5	10.0	9.9	19.3	49		2.54
6000	110	18.00	18.0	14.3	9.0	20.5	50		2.44
6000	120	17.10	16.2	14.1	8.2	17.5	49		2.80
7000	140	25.80	9.0	20.4	5.4	20.9	40		1.90
Average						17.0	47		

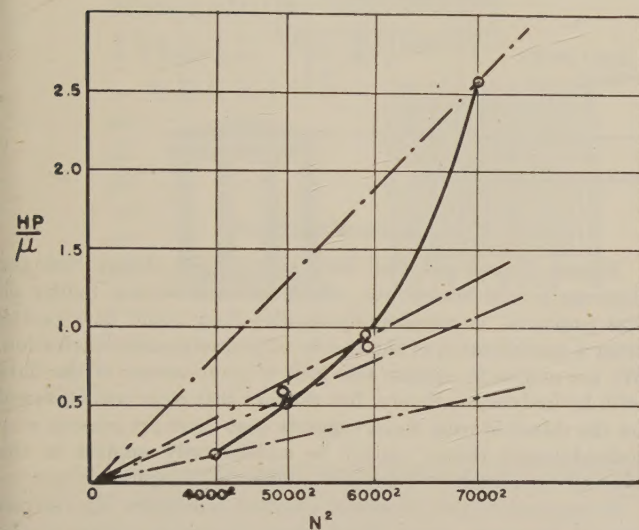
^a Revision of author's Table 5.


FIG. 11

ections and therefore these losses would be of a very different order from those tabulated.

The data plotted in Fig. 8 of the paper do not appear to correspond completely to those listed in Table 5, but cover many more observations. The general distribution of these observations appears to justify drawing two straight lines similar to those shown, with a spindle-speed exponent of approximately 2, which might be interpreted as a statistical proof of Equation [1]. However, this interpretation would be valid only if these lines represented the power-speed relationship for two definite viscosities. Inspection of the chart will show that this is not so, as points covering a wide range of viscosities fall in the same group. There is also a very great spread in power values for similar viscosities at the same speed.

In view of the foregoing reasoning, it would seem that we should look for some other explanation of the apparently high idle-power losses. One possible explanation is indicated by the author's statement that the power values presented were converted from measured input power by means of data obtained from motor-efficiency tests conducted by the manufacturer of the motor. As the particular motor used in these tests was entirely special, and as the rotor could not be assembled on its shaft and bearings until installed in the machine, after which an efficiency test on the motor alone would be difficult, it is questioned whether the efficiency data were taken from tests on some other motor. If we assume that the actual efficiency values of the special motor in the low-power range were less than the assumed values, this might well account for the apparent high values of idle power, and also the apparent small decrease in power with viscosity indicated in Table 7.

In any further investigations with this apparatus it would seem highly desirable to introduce a calibrated and graduated spring

coupling of high elastic modulus between the motor shaft and spindle. By means of a low-power microscope with stroboscopic illumination, accurate measurements of torque may readily be made, even with a relatively rigid coupling. This will permit the checking of all power readings previously obtained and provide a more precise method of ascertaining the efficiency of metal removal.

A minor correction would seem to be in order, where it is stated that the "motor bearing shoes are of the fixed type." This statement might lead to some confusion of thought as actually both the motor bearings and spindle bearings are of the tilting-pad type. Presumably the term "fixed" is intended to imply only that the radial adjustment of the motor shoes is set at a fixed value, not that the shoes themselves are incapable of tilting.

We are glad to note that the author is continuing these investigations with a transverse feeding mechanism arranged to permit tests with high-speed milling cutters. These tests should provide useful information concerning the power required for milling, at high speed and at high rates of stock removal, the various aluminum alloys now utilized in industry.

W. W. GOEHRING.⁷ The many factors involved in developing this high-speed lathe present a challenge to all of the machine-tool manufacturers and their suppliers. As manufacturers of ball and roller bearings, we are specifically interested in the problems connected with the bearings on the spindle.

It was pointed out that as much as 56 hp might be required to drive the spindle at 9000 rpm on "Filmatic" bearings when a cut was not being taken. This is considerable horsepower to transfer to heat and take out again. In an attempt to reduce this idle horsepower, the clearance of the radial and thrust bearings of the spindle was increased, and a lighter oil was used in the lubricating and hydraulic systems. Increasing the clearance of the bearings had very little effect on the idle horsepower and the use of the lighter oil resulted in an average reduction of approximately 17 per cent, which was not very much.

We would like to know what effect the change in bearing clearance and oil viscosity had on the rigidity of the spindle. From long experience in the machine-tool field, we are conscious of two fundamental spindle problems, namely: (1) keeping the torque and resultant temperature to a minimum; and (2) eliminating internal clearances of the radial and thrust bearings to maintain maximum rigidity, which is of prime importance in obtaining accuracy and finish.

With the ball-and-roller-bearing spindle, we are able to maintain rigidity independent of speed and oil viscosity and pressure. In contrast to the 56 hp required to drive the Filmatic spindle at 9000 rpm, only 24 hp to the motor is required to drive the ball-and-roller-bearing spindle shown in Fig. 12 herewith at 10,000 rpm, under 225 lb radial load and 200 lb thrust load, with 2 gal of DTE heavy-medium oil per min passing through each head.

⁷ Senior Division Engineer, SKF Industries, Inc., Philadelphia, Pa. Jun. ASME.

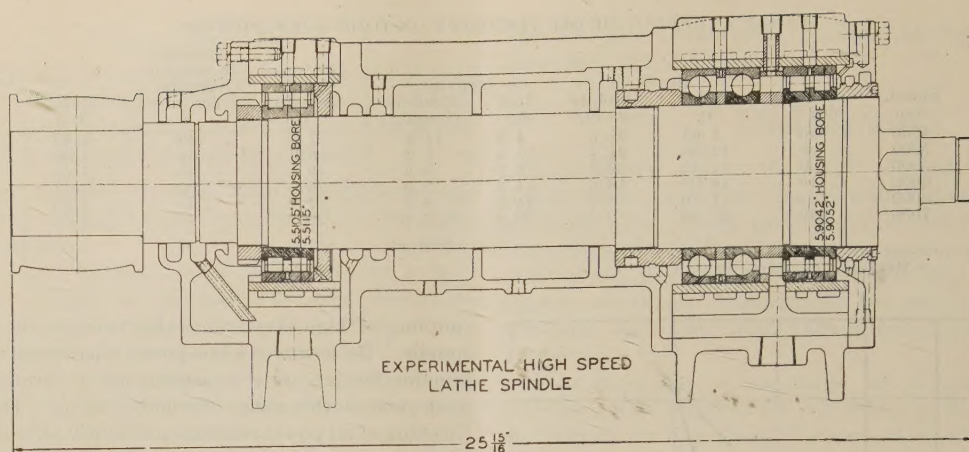


FIG. 12 EXPERIMENTAL HIGH-SPEED LATHE SPINDLE

An experimental spindle, such as shown in Fig. 12, was run for more than a year, during which time we observed more than 100 different test runs under varying conditions. The spindle was bolted to a lathe bed and belt-driven by a 15-hp, 1800 rpm, 220-volt 2-phase motor with a jackshaft and two step-up belt drives. A 225-lb radial load and 200-lb thrust load were applied for all runs through a deep-groove ball bearing at the work end of the spindle.

Three methods of lubrication were tried, namely, oil level, oil mist, and circulating oil. At 10,000 rpm, it was necessary to use circulating oil, and in addition to radiation to the air and conduction through the machine base, the heat was dissipated by means of cooling the circulating oil. Oil-mist lubrication as used was unsatisfactory over a speed of 3600 rpm, as it failed to carry off sufficient heat.

The tests have shown that 4-in.-diam-bore bearings, whether roller or ball, can be operated without difficulty at 10,000 rpm, and that the temperature of the spindle will depend upon the arrangements for cooling and circulating oil. Two gallons per minute to each head of DTE heavy-medium oil, which has a viscosity of 320 SSU at 100 F are sufficient for cooling and yet not enough to create excessive torque.

A great number of tests were run with a very light oil for the purpose of generating as little heat as possible. It was found, however, that the low-viscosity oil at relatively high operating temperature did not have sufficient lubricating ability, thus causing some wear on the rollers at the end of the cage prongs of our standard cages at 10,000 rpm. By redesigning the cages to cover the entire length of the roller for high-speed operation and by using an oil of sufficient viscosity, the problem of wear is reduced to insignificance.

We found that water circulated through jackets around the outside diameter of bearings is not a satisfactory cooling method as it cools the outer more than the inner, thereby altering bearing fit. Therefore this medium of cooling was given no further consideration in our tests, with the result that the housing design can be simplified considerably.

AUTHOR'S CLOSURE

Messrs. Ernest and Dall have shown quite clearly that the decrease in idle horsepower, which resulted when a lighter oil was employed, is considerably smaller than would be expected from a consideration of the theory of hydrodynamic lubrication. We are unable to explain this lack of conformance of the data with hydrodynamic theory, but the fact that wear was observed on the thrust-bearing shoes suggests that there are reasons why hydrodynamic theory cannot be satisfactorily applied in the over-all consideration of the performance of the spindle.

In response to the discussers' question concerning the conversion of measured motor input power to motor output at the coupling, we should like to explain that the conversion was not arbitrary in any respect. The motor manufacturer conducted the efficiency tests on the spindle drive motor after it was installed on the lathe. The motor shaft coupling was disconnected and we are assured that the efficiency data which were obtained and used are reliable.

Mr. Goehring questions what effect the change in bearing clearance and oil viscosity had on the rigidity of the spindle. Although we did not make any specific tests to check this point, we observed no decrease in the accuracy of the cuts or the quality of the machined surfaces and, therefore, conclude that the spindle maintained sufficient rigidity for this application. The results of the tests covering the performance of an experimental ball-and-roller-bearing spindle, reported by Mr. Goehring, are important in that they indicate the ability of relatively large antifriction bearings to operate satisfactorily at 10,000 rpm.

Since the paper was presented, the author's company has changed the temper designations for aluminum alloys. The old temper designations appear in the paper, but the equivalent new temper designations are presented here:

Old temper designation	New temper designation
14S as-rolled	14S-F
14S-T	14S-T6
24S-T	24S-T4